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PAPERS PRESENTED AT THE FIRST ANNUAL CONFERENCE ON AIR CONDITIONING

HELD AT THE

UNIVERSITY OF ILLINOIS

MAY 4 AND 5, 1936

CIRCULAR No. 26

ENGINEERING EXPERIMENT STATION

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I. WHAT IS AIR CONDITIONING?

Arthur C. Willard*

This is a *non-technical conference* for the non-technical person interested in air conditioning for human comfort. No attempt will be made to discuss the many phases of *industrial air-conditioning* so important in the manufacture of many commercial products, such as cotton, rayon, celluloid, candy, tobacco, color printing and others.

Air conditioning is *not a new discovery* but the application of principles long known to the heating and ventilating engineer and the physiologist to the problem of keeping people comfortable by artificial means through the proper control of the atmospheric environment. Moreover, air conditioning in this latitude should provide for *a year round service,* and hence we must be prepared to deal with both winter and summer conditions.

Briefly stated, (1) a complete system of air-conditioning for *winter service* should provide for the *controlled* production of *heat* and *humidity,* with the *removal* of a large part (75 per cent or more) of *dust* and *dirt* floating in the air; and (2) a complete system of air conditioning for *summer service* should provide for the control or regulation of air temperature or of humidity or of both within certain rather wide limits by either *air cooling devices* of various types or by *dehumidifiers* or by both. By reducing the humidity, that is, *condensing* or *absorbing* the *moisture* in the air, the amount of actual air cooling may be greatly reduced. In any case, air cooling should be moderate in amount.

In the intensive pursuit of an objective it sometimes happens that we become so engrossed in developing the means for accomplishing the objective that we forget or lose sight of some of the vital factors inherent in the objective. I think this has happened and will continue to happen in our pursuit and development of the means employed in modern air conditioning for human comfort.

Our sole purpose and our ultimate objective in the development of both winter and summer air conditioning is *to make human beings comfortable.* We are not interested essentially in temperatures, humidities, air motion, dust and bacteria counts, and other indices of air conditions, but in human comfort. We know today that this involves, during much of the year in this country, the maintenance of, first, a proper atmospheric environment, and, second, a proper physical environment, both of which must be agreeable to the occupants. The

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former depends largely on operating equipment, while the latter depends on the structure or building surrounding the occupants who are to be made comfortable. Up to the present, much has been done and great interest has been shown in the equipment, while the reverse is generally true in the case of the structure. In my opinion, the second factor is fully as important as the first, and may completely determine the type, the size, or capacity (including cost), and the operating cycle of the equipment, as well as the operating charges. I shall endeavor to develop my reasons for this opinion as we proceed.

Human comfort is an elusive thing to measure and to specify, as well as to achieve. It means, in terms of the usual indices of temperature, humidity, air motion, and others of which I have already spoken, one thing for you and another for me. But for all of us, and this is an unfailing criterion, it is successfully accomplished when we are unconscious of the necessity for any physiological adjustment to our environment, atmospheric or otherwise.

By physiological adjustment I mean such physical reactions as perspiring and shivering, to take extreme cases. In between these extremes we frequently experience a rapid increase or decrease in blood circulation through the blood vessels of the skin, especially those of the head, the hands, and the feet. When these physical reactions are violent enough to attract our attention we say we are "too hot" or "too cold." But when these physical reactions are so moderate and gradual as not to attract our attention we say we are quite comfortable. In other words, so long as the heat control mechanisms of the body operate insensibly without any knowledge on our part of their functioning we are comfortable and satisfied both winter and summer. Therefore, we may say that human comfort prevails whenever the heat loss from the body is exactly balanced by the heat production of the body, and we are not conscious of any physiological adjustment in order to maintain such a balance. It is hardly necessary to point out that, in any case, a perfect balance must exist between the heat production and the heat loss of the human body, or discomfort, distress, and even death itself may result. We may be most uncomfortable while the balance is being maintained.

Possibly it has never occurred to you that such a heat balance has to be maintained—in fact very accurately maintained—or death will soon result. The facts are that in order to live we all generate heat within our bodies at a very definite rate as a result of the oxidation of human tissue by the oxygen carried by the red blood. The oxidation products such as carbon dioxide and water are carried away by the venous blood and exhaled from the lungs. But most of the heat generated in the process has to be dissipated or lost from the skin itself. In other words, this heat flow takes place always in only one direction, that is *from* the body to the surrounding air and the visible wall and glass surfaces. Moreover, it must proceed at such a rate as to exactly balance the heat produced in each individual—not too fast and not too slow.

In promoting human comfort by producing and controlling those conditions under which this balance will take place unconsciously, the air conditioning engineer has devoted his energies to the factors on the heat *loss* side of the equation since there is very little he can do on the heat *production* side. Hence he has made elaborate studies of the way in which heat is lost from the body by radiation, convection, and evaporation under various atmospheric conditions in which the laboratory air and walls are at the same temperature. But in actual practice the laboratory conditions are seldom duplicated, and our laboratory indices may not fit the actual case in hand very well.

All I am trying to say is that the indications of all the instruments in the world can never take the place of the sensations of a human being when it comes to the determination of whether or not that human being is comfortable or otherwise. Moreover, and this is the essential point, human discomfort is never registered by any individual until he or she becomes conscious of the fact that a physiological adjustment is being made in order to balance body heat loss against body heat production. These adjustments involve the surface condition of the body such as the temperature of the skin and the amount of perspiration present. Clothing may assist or interfere with the adjustment, but does not change the essential phenomena in any way. Since body heat production may range from about 400 B.t.u. per hour for man at rest to three or four times this amount for man at hard labor or exercise the heat control mechanism of the body must be extremely flexible in operation. (Note: It requires 144 B.t.u. to melt one pound of ice at 32 degrees Fahrenheit.) The heat control mechanism of the body functions by rapidly and effectively increasing or decreasing skin temperatures by more or less blood circulation, and skin moisture by more or less perspiration, and even by involuntary muscular activity such as shivering to increase heat production.

Since we are concerned only with *surface heat loss* phenomena, such as radiation, convection, and evaporation, the laws of which are well known, we must determine which, and to what extent each, of these phenomena takes place when human comfort prevails. For conditions of rest the heat dissipation amounting to about 400 B.t.u. per hour for an average adult is divided approximately as follows: radiation 210 Bt.u., convection (still air) 90 B.t.u., evaporation 100 B.t.u. (without sensible perspiration). Hence our problem is to create and maintain an *atmospheric* and *structural* environment in which the body heat losses *will occur* by the methods and in the amounts just indicated. It is apparent that radiation which is quite independent of the atmospheric environment but wholly dependent on the structural environment disposes of *more than half* of the body heat production when the individual is at rest and comfortable.

Hence, as I indicated at the outset, we should give far more attention than in the past to the construction and insulation of the walls, floors, ceilings and windows of our houses, when we are planning to air condition them for human comfort. Such attention is even more important for successful winter air conditioning than it is for summer air conditioning, and the air conditioning system of the future must provide for both conditions, or all year round service.

II. COMFORT CONDITIONS AND PHYSIOLOGICAL FACTORS IN AIR CONDITIONING

F. C. HOUGHTEN*

As far as most of us are concerned, air conditioning represents a new development in applied science and industry. These newer developments with which we are acquainted involve the application of air conditioning for human comfort and health. Air conditioning has, however, been used for several decades in industry to make the working of certain materials more satisfactory. In this connection, air conditioning is, and has been, necessary for the satisfactory rolling of cigars, spinning and weaving of textiles, the drying of macaroni, and more recently, in the making of candy and many other processes. These applications are known as industrial air conditioning, as contrasted with comfort air conditioning.

The recently developed popular conception of as well as the present rapid growth in air conditioning has to do with comfort air conditioning applied to theatres, churches, auditoriums, schools, restaurants, homes, etc., in order to improve human comfort and health. Air conditioning for comfort and health has an entirely different appeal from that of most other branches of applied science or engineering, and industry, since its sole purpose is human comfort and health. The underlying requirements are knowledge concerning the human body and how it reacts to the atmospheric environment in which it exists. Most of us seldom give any consideration to the air which surrounds us; we simply accept it as one of the few remaining free requirements of life. However, if we examine our relation to the air, we find it of profound significance in our daily life.

We take into our system, at most, a very few pounds of food and water during a day, yet we give a great deal of attention to regulations governing its production, distribution, and preparation in order to insure our health. With little thought we take from 25 to 30 pounds of air per day into our respiratory tract, consisting of the most sensitive organs of the body. Further, our bodies are at all times bathed in this same air, which in a large measure controls the rate of heat production in and dissipation from our bodies through the processes of life spoken of by the physician and physiologist as 'metabolism.'

Viewed from this point of view, our atmospheric surroundings which we take so much for granted become of great importance, and

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greater attention to factors affecting the purity and satisfactory condition of the air may be expected to result in a profound effect on our comfort, health and longevity.

During the past fifty years the average span of human life has been greatly increased, largely through the development and dissemination of knowledge concerning the sanitary control of the food and water supply, including the proper disposal of sewage. As we examine the facts we find that this improvement in health and increase in the length of life has resulted in a great measure from the control or elimination of diseases of the digestive tract, and as a result we find today that the outstanding ailments are diseases of the respiratory tract. With similar vigilance applied to the control of the air we breathe, and in which our bodies are at all times bathed, through air conditioning, it is not too optimistic to anticipate a similar increase in the average span of life through the control of diseases of the respiratory tract. In this respect, air sanitation or air hygiene are terms synonymous with air conditioning, whose true significance might be better understood. Thus viewed, air conditioning is an important factor in our daily life, and any man connected with its development may view his calling as a noble one.

Air conditioning may be defined as the control of any or all of the physical or chemical properties of the air in some enclosure for a purpose. Air conditioning for health and comfort involves the control of those properties of the air which affect comfort and health. And a knowledge of these physical or chemical properties and their effect on the human system is absolutely necessary for a proper understanding of comfort air conditioning. To practice air conditioning, or to be engaged in the installation of air conditioning equipment in buildings, a man must not only be a good engineer, but he must also have a comprehensive knowledge of physiology and the physiological reactions of the body to its atmospheric environment.

The properties of the air, or the air conditions which affect comfort and health are, in the order of their importance, (1) the temperature of air, (2) the moisture content of the air, (3) the motion of the air, (4) the dust content of the air, (5) the freedom of the air from odors, (6) the freedom of the air from other harmful vapors, gases, bacteria, or other harmful substances. The temperature, moisture content, and motion of the air, combined, determine the personal feeling of warmth; or, in other words, these three qualities of the air determine the rate at which heat is given off from the body to the surrounding air under any given condition of clothing and activity.

We are all fairly well acquainted with the effect of the temperature of the air itself, that is, the temperature of the air as given by an ordinary dry-bulb thermometer. We know that when the air temperature is somewhere around 70 deg. F. we are comfortable, when seated at rest and normally clothed. We also know that as the temperature of the air falls we feel an increasing sense of coolness, while as the temperature rises we feel an increasing sense of warmth. Usually to a lesser extent, the moisture content of the air also affects our sense of warmth. We are most cognizant of this effect in the heat of the summer, when we are all accustomed to the expression "it is not the temperature, but the humidity or moisture content, which makes us so uncomfortable." A range of 95 to 110 deg. F. air temperature may be endured without great hardship in Arizona, where the air is very dry, but these same temperatures with the high moisture content found in Illinois become unbearable. The difference is due to the moisture content of the air.

Likewise, most of us have a fairly clear conception of the effect of air movement on one's feeling of warmth, or in other words, the cooling effect of wind or air movement. We are less cognizant, however, of the profound cooling effect resulting from lower velocity air movements, such as pertain in any well-ventilated room. The movements constitute the small eddy currents which cause the rapid dispersion of a puff of smoke.

We have mentioned the temperature, moisture content, and movement of air as affecting the feeling of warmth. It is interesting to go more deeply into this subject and see just what effect they really do have. The human body is frequently likened to a locomotive, or other prime mover. Instead of coal and draft, we require food and oxygen, which through oxidation supplies the necessary heat to maintain the rigidly required body temperature of about 98.6 degrees, and the necessary energy for the various processes of life and other forms of work.

The processes of life of an average-sized man, 5 feet 9 inches in height, and weighing 160 pounds, require a minimum production of about 400 heat units, or British Thermal Units of heat, per hour, for moderate temperatures of from 65 or 70 to 80 or 85 deg. F., with moderate relative humidity and with moderate exercise. Smaller persons produce less heat. This amount of heat must be dissipated from the body to the atmosphere either in the form of radiated heat, convected heat carried away by air currents, or latent heat of evaporation of moisture, either from the respiratory tract or perspiration from the body surface. In order to maintain a constant temperature of 98.6 deg. F., the rate of heat dissipation from the body must exactly equal the rate of heat production within the body. For colder conditions this equilibrium is maintained by the production of more heat within the body, unless the cooling effect is counteracted by applying more insulation in the form of clothing. Naturally, a more comfortable method of maintaining this temperature equilibrium is to maintain a proper atmospheric condition, or air temperatures, so that this heat may be dissipated without undue effort.

For higher air temperatures the rate of heat dissipation to the air by radiation and convection naturally decreases in accordance with the laws of physics, and the only recourse which the body has to avoid a rise in body temperature above 98.6 deg F. is to increase the rate of heat dissipation by evaporation. This is accomplished by the body (through some form of thermostatic control) by making available more moisture in the form of perspiration for evaporation. As the atmospheric temperature rises from somewhere around 70 to about 80 deg. F., with a 50 per cent relative humidity, the body is able to increase heat loss by evaporation without noticeable or sensible perspiration. At 70 deg. F. dry-bulb, approximately 26 per cent of the total loss of 400 B.t.u. per hour is by evaporation, while for the higher temperature of 80 deg. F. mentioned, 44 per cent is lost by evaporation. At 20 per cent relative humidity the body may lose as high as 60 per cent of its total heat loss of 400 B.t.u. per hour by evaporation without sensible or noticeable perspiration. However, as the temperature rises above these limits, the body, in attempting to increase evaporation, makes perspiration available in quantities which are sensible and uncomfortable. Any atmospheric condition in a hot climate meant for comfort must therefore make possible the loss of sufficient heat by evaporation of moisture from the body surface, without sensible perspiration. This can be accomplished either by keeping the dry-bulb temperature sufficiently low or by controlling the moisture content of the air.

These facts make apparent the need in air conditioning of maintaining a proper balance between the temperature of the air and its moisture content, and it becomes obvious that control of the dry-bulb temperature of the air alone is not sufficient. Extensive research during the past decade has resulted in knowledge concerning the proper relation between temperature and moisture content of the air for comfort, and there has been established an "effective temperature scale," taking these two factors, as well as air motion, into account. This scale, known as the "effective temperature scale," must be used in air conditioning for human comfort. As an example, for winter heating a 66 deg. "effective temperature" with a tolerance of about two degrees, has been found to be comfortable to most people. This standard of 66 deg. effective temperature may be had with 66 deg. dry-bulb and 100 per cent relative humidity, 70 deg. dry-bulb and 50 per cent relative humidity, or 74 deg. dry-bulb and 10 per cent relative humidity.

With the increased application of summer cooling as a factor in air conditioning it has become apparent that people become partially acclimated to outdoor temperatures, and due to this acclimatization they desire a higher temperature for comfort in the summer than they do in the winter. As contrasted with 66 deg. effective temperature for comfort with winter air conditioning, about 73 deg. effective temperature is desirable for summer air conditioning. This may be had with 73 deg. dry-bulb and 100 per cent relative humidity, 78 deg. dry-bulb and 60 per cent relative humidity, or 84 deg. dry-bulb and 20 per cent relative humidity. Very high and very low relative humidities are apparently to be avoided in either winter or summer air conditioning for human comfort.

Increased air motion, as we all know, serves to make one feel cooler at the same temperature, and may be satisfactorily used in many instances in producing comfort cooling in summer air conditioning. However, the possibilities of making use of air motion are limited to such velocities as do not annoy one in any particular occupation.

Sufficient knowledge of the Effective Temperature Scale and its application to comfort air conditioning cannot be gleaned from a talk of this kind. More comprehensive discussions of the subject, as well as other subjects pertaining to air conditioning, may be found in the Guide of the American Society of Heating and Ventilating Engineers, as well as in several other available text books covering the subject.* An instrument necessary to take proper cognizance of the factors entering into the Effective Temperature Scale is a sling psychrometer, which gives both the dry-bulb and the wet-bulb temperature of the air, and from which the moisture content or relative humidity may be determined.

Of slightly lesser importance in air conditioning for human comfort is the cleanliness of the air as regards dust, odors, and other harmful substances. The atmosphere in most of our cities is heavily laden with dust, which is not only damaging to furnishings in our homes, but is also probably harmful to our respiratory tract. At least, we can say that the presence of dust in the air is highly unpleasant. Air-borne dust may be removed by several means, including washing

*A reproduction of the A.S.H. and V.E. Comfort or Effective Temperature Chart will be found on p. 94.

the air with a spray of water in an air washer, or by filtering the dust out of the air with any one of a number of forms of commercial air filters. Air cleaning is always an essential part of air conditioning in urban communities.

It costs money to condition air, and, therefore, there is a marked tendency where conditioned air is supplied to reduce the quantity of outside air to a minimum. This tendency has resulted in an extensive study during the past two decades of the harmful effect on the human system of breathing recirculated air where too small a quantity of outside air is supplied per person. We probably all remember warnings of the dire results of rebreathing carbonate in closed or poorly ventilated rooms occupied by groups of people. Recently it has been proven that carbon dioxide is not the controlling factor under such conditions. The presence of odors in the air resulting from persons occupying the space more often determines the amount of outside air necessary, in order to reduce such odors below the threshold of our senses. Recent work has shown that from 10 to 20 cubic feet of air per person per minute is necessary in most cases to avoid odors. The exact amount of outside air required in a given space depends upon the cleanliness of the occupants, the temperature maintained, and the degree of freedom from odors desired. This is an important factor in school ventilation. A few years ago, when carbon dioxide was considered to be the determining factor, most of our States enforced laws requiring 30 cubic feet of outside air per person per minute in schoolrooms. Additional knowledge has in recent years reduced this requirement to from 15 cubic feet per person per minute in some instances, to as low as 10 cubic feet in others. The same standards should pertain in other occupied spaces. Other harmful or obnoxious substances in the air are seldom met with except in industrial establishments. A simple requirement in this connection is that air for human occupancy should be refreshing and free from any sense of obnoxious odors or harmful substances.

In air conditioning for human comfort it must be constantly kept in mind that the sole purpose of the application of air conditioning is to give the individual a pleasant, healthful, and comfortable surrounding. Any such annoyance as undue noise from the equipment, or even the sight of unpleasant equipment within the occupied space, may entirely defeat the advantages otherwise gained by making the occupant conscious of the presence of the equipment to the extent that it becomes annoying. This fact must be constantly kept in the foreground in any application of air conditioning for human comfort.

III. AIR CONDITIONING AND ITS EFFECT ON HAY FEVER AND POLLEN ASTHMA

WILLIAM H. WELKER*

The number of persons in the United States suffering from hay fever has been estimated at from one to four million by different men working in the field. It is generally agreed that there has been a marked increase in the number in the last decade. This is explained on the basis of increased exposure to pollen for which automobile transportation is largely responsible.

A little over half a century ago it was first definitely shown that hay fever was caused by pollen. It required more than 25 years to establish the fact that the pollens of common and giant ragweeds are responsible for a very large percentage of the existing hay fever. Golden rod, which had received the blame of the public, is found actually to be responsible for such a small percentage of hay fever that it can be disregarded as a causative agent. Its pollen is sticky and is consequently not carried by the wind to any appreciable extent. The ragweed pollens are believed to be responsible for approximately 85 per cent of the autumnal hay fever in that portion of the United States east of the Rocky Mountains. In the Pacific and Rocky Mountain States the wormwoods are the leading causative factors. Other plants, grasses, and trees likewise give off pollens which produce somewhat similar effects but their pollenating seasons are very much shorter, and, consequently, the disturbance lasts for only a few days. The ragweed season is much longer. In the Chicago area it begins about the 15th of August and lasts until the first killing frost.

Billions of microscopic grains of pollen from the ragweed, or other plants, are dislodged from the plant and carried by the air currents. Because they are so light they may be carried for great distances through the air. There are observations showing that relatively large quantities of certain pollens have been carried as far as 35 miles, and, in a few rare instances, it has been found that they were definitely carried over 200 miles. Pollen grains have also been found in the air as high as 17 000 feet above the ground. From these observations it can be seen that the air currents are quite effective in distributing the pollen over relatively large areas. The pollen grains are so small that they cannot be seen with the naked eye. (Size of pollen grains of common ragweed, 15 microns; of giant ragweed, 20 microns; one

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FIG. 1. GIANT RAGWEED PLANT

micron = one thousandth of a millimeter.) They are yellow in color and are responsible for this color as seen on the ragweed plant during the pollinating season.

The symptoms of hay fever are running nose, watery and itching eyes, and general symptoms accompanying a typical head cold. Fortunately, these symptoms appear only in those individuals who have become sensitized to the pollen proteins. Years ago by animal ex-

FIG. 2. GIANT RAGWEED PLANT DURING POLLINATING SEASON

perimentation it was conclusively proven that the individual can be sensitized to a given protein by injection of this protein into the blood-stream, provided the protein is antigenic. If a second injection is made after an interval of a week or ten days, the animal shows symptoms that are quite characteristic. The quantity of protein required in the first, or sensitizing, injection is so small that it is almost unbelievable.

FIG. 3. POLLEN GRAINS OF GIANT RAGWEED PLANT
(Magnified 600 diameters—with dark field illumination)

As a result of these findings it seems obvious that persons reacting to pollen must have had some of the protein of the pollen enter their bloodstream. Apparently in some individuals, and perhaps under conditions where there is congestion of the lining mucous membrane, these proteins may be absorbed from the air passages and the lungs. If subsequently an appreciable amount of pollen lodges in the air

Fig. 4. Short Ragweed Plant

passages and the lungs, sufficient protein may be absorbed to produce the symptoms characteristic of hay fever.

The only relief offered by medical treatment consists of a series of injections of pollen protein in increasing amounts two months or so before the hay fever season starts in an attempt to desensitize the individuals to pollen protein. Some allergists recommend injections

FIG. 5. SHORT RAGWEED PLANT DURING POLLINATING SEASON

over the whole year. If this treatment is successful, the individual who formerly showed symptoms of hay fever on exposure to pollen fails to show such symptoms, or shows symptoms of very much lesser degree. Not all individuals who usually suffer from hay fever respond to this treatment, and those that do respond must be treated each season because the so-called desensitization lasts only a relatively short period of time. Occasional exceptions to this rule have been noted.

For four hay fever seasons in collaboration with Dr. B. Z. Rappa-

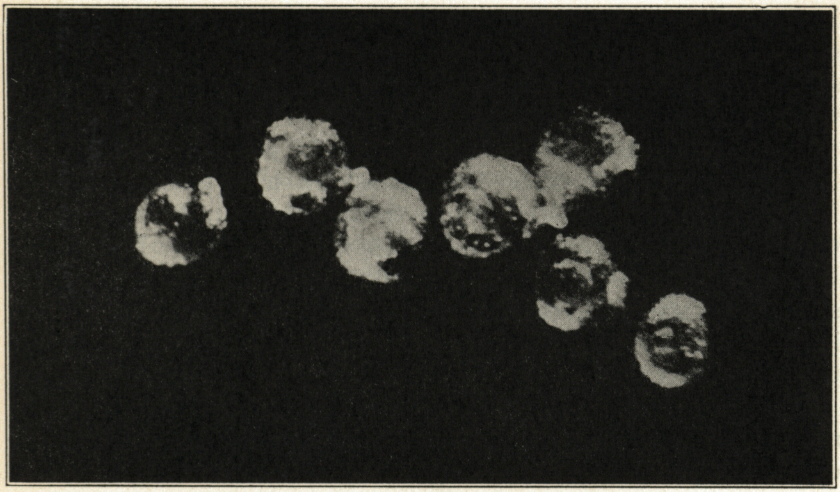

FIG. 6. POLLEN GRAINS OF SHORT RAGWEED PLANT
(Magnified 600 diameters—with dark field illumination)

port and Dr. Tell Nelson of the Department of Medicine studies have been conducted on the effect of filtration of air on the relief of symptoms in hay fever and pollen asthma. Our experiments were undertaken to see whether, by relatively simple apparatus, we could remove a sufficient portion of the pollen so as to give relief to hay fever sufferers. We used a six-bed room the first season, an eight-bed ward the second season, and an eight- and a four-bed ward in the third season. One hundred and seventeen patients with hay fever of the pollen type, and with skin tests positive to ragweed pollen protein, were studied for an aggregate of 483 nights. The air was propelled from the outside into the ward by means of centrifugal blowers. This arrangement places the ward under a slight positive pressure and consequently no particular precautions had to be taken as to sealing the windows and doors. The filter material that we have found most successful is a cellulose filter (really a paper) of very loose texture, through which the air is drawn. As soon as these filters show any clogging effects from the dust that accumulates on them, they are replaced with fresh filters. These filters are inexpensive.

The amount of pollen in the air was determined by counting the number of pollen grains on definite areas of microscopic slides. These slides were coated with a thin film of vaseline, which held pollen that came in contact with it. The slides were exposed in still air (settling count) and in moving currents of air for 24 hours. During one of the hay fever seasons in our location in Chicago, the maximum was slightly below 700 pollen grains per cubic yard as determined by the settling counts. By comparing the pollen counts outdoors and in the ward we found that we could remove between 98 and 99 per cent of the pollen. This is sufficient removal to make practically all the hay fever sufferers comfortable. To attain such high efficiency it was found necessary to have the air pass through the filters at slow speed; this means that the filter areas have to be relatively large. Mechanical vibration has to be eliminated from the filters, since this produces a sifting effect and has a tendency to permit pollen to go through. This is accomplished by making flexible connections between the blower and the filter forms.

In the air from which practically all the pollen had been removed, most of the hay fever sufferers were relieved completely, or almost completely, of all their symptoms in from one-half to one-and-a-half hours. Freedom from these symptoms continued as long as they remained in the pollen-free air. On their return to pollen-laden air, either outdoors, or even in the hospital corridors, the symptoms rapidly reappeared. Our experiments showed that personal efficiency could be increased by providing filtered air in the sleeping quarters of hay fever sufferers. Under these conditions they secured a good night's sleep, which was not usually the case if they were exposed during the night.

Individuals who have suffered from hay fever for a number of seasons may develop asthma for which the pollen is solely responsible. The number of hay fever sufferers in which asthma ultimately develops has been given by various authorities as ranging from 15 to 40 per cent.

Pollen asthma is a more serious disturbance, accompanied by changes in the lungs, and, therefore, requires longer confinement in filtered air before the relief from symptoms is obtained. Forty-seven pollen asthma patients were studied in this work. Our patients required from a day-and-a-half to four, or even five, days continuous confinement before the symptoms disappeared, and in a few of the cases relief from symptoms was not obtained even then. For some of them the relief was profound. They were unable to sleep in pollen-laden air unless they could do so sitting up in a chair. In the pollen-free room they could get normal sleep.

The results in these experiments show definitely that when pollen is excluded from the environment of individuals suffering from hay fever, or pollen asthma, their symptoms disappear, and they remain free from symptoms as long as they are not again exposed to pollen-laden atmosphere.

We feel that our findings can be utilized in the treatment of hay fever and pollen asthma in the home and elsewhere, in the form of an apparatus which will deliver into the room sufficient air for the occupant and at the same time carry it through filter material that will effectively remove the pollen. It would be applicable as well to hotels, factories, schools, stores, and places for public assemblies. In fact, there are at the present time certain stores and public buildings equipped with filters of the same type as we have been using in our experiments that have been discovered by the public to give marked relief to hay fever sufferers. These buildings have become literally hay fever resorts. The removal of pollen in these buildings is not, of course, as complete as we have obtained in our experiments here in the Illinois Research and Educational Hospitals because of the less perfect design of the filters. There are two private homes in the State of Illinois that are completely equipped for filtration of the air for the removal of pollen which have given very definite relief to the hay fever sufferers in these homes. The question of the efficiency of the pollen removal necessary to give relief is dependent very largely upon the degree of sensitivity of the hay fever patient. If the sensitivity is great, high efficiency is essential; if the sensitivity is of a low grade type, less efficient filters will give relief.

We have studied the effect of air cooling without removal of any of the water from the air. This results in a higher degree of saturation of the air with water. In our experiments with a 10 deg. cooling below the temperature in the rest of the hospital, we had an average increase of 10 per cent in the relative humidity. Our hay fever and pollen asthmatics were definitely less comfortable in this cooler room, with the increased percentage saturation of water, than they were in rooms with higher temperature but lower percentage of water saturation.

We also studied the effect of the air with low relative humidity on a series of pollen asthmatics. This ward with eight beds was run at around 30 per cent relative humidity and a temperature of 79-80 deg. F. This condition seems to be slightly more favorable for pollen asthmatics than a room that had a higher relative humidity. This study was conducted because it was observed in the ward where the humidity was not controlled that after a heavy thunder storm all of the patients had asthmatic seizures. In this practically pollen-free air the disturbance could not be attributed to pollen. In the following hay fever season an experiment was conducted to see whether these seizures would occur where the relative humidity was maintained at a constant point. It was found that they do occur, but the onset is delayed, the attacks are possibly not as severe, and recovery seems to be somewhat more rapid. However, it is quite clear from the results of these experiments that the prime reason for the asthmatic seizures is not the marked increase in the relative humidity, which occurs as a rule during heavy thundershowers. Since ozone is an accompaniment of heavy electrical storms, we introduced ozone into the air of the ward in quantities larger than occur during electrical storms without producing any effects whatever. We have also made preliminary experiments on the effect of negative ionization of the air with no effect whatever on the patients in the ward. We feel that these experiments should be repeated using very much higher concentrations of negative ions. The reason for the onset of asthmatic seizures in a pollen-free atmosphere under these conditions remains undetermined.

From our results it is apparent that the important factor in air conditioning, so far as relief of symptoms in hay fever and pollen asthma is concerned, is the filtration of the air in such fashion that the major portion of the pollen is removed. Additional comfort can be secured by cooling with dehydration.

If air filtration for the removal of pollen is to be used for the relief of symptoms in hay fever, or pollen asthma, it is desirable that a member of the medical profession be called to diagnose the case. Such air conditioning is ineffectual in other conditions. It is desirable, therefore, to be certain that pollen is the causative factor in cases where air filtration is to be utilized as the relief measure.

A number of manufacturers are marketing pieces of apparatus that can be connected to the window of a room by which the air is drawn in from the window and filtered through filter material. Any such machine that does a satisfactory job of removing practically all the pollen and supplying a sufficient volume of air will be capable of giving relief to hay fever and pollen asthma sufferers. Houses equipped with hot-air heating plants, and particularly those with forced circulation, can be equipped with filters for the removal of pollen with a minimum of expense.

In purchasing an air filtration machine, there are certain points that can be determined by direct observation. First, the area of the filter material ought to be relatively large. This will insure the flow of air through the filter material at a low speed, which is essential if a high degree of efficiency is to be maintained. Second, there should be no appreciable vibration in the apparatus; should this occur, there is a danger of shaking the pollen through the filters. Third, the volume of air delivered by the machine for a given unit of time must be sufficient to keep the individual comfortable. The Laboratory of the Council on Physical Therapy of the American Medical Association in Chicago has studied the efficiency of most of the air filtration machines for the relief of hay fever, and the public can secure information on this point by contacting that organization.

Our work suggests the possibility that efficient air filtration with recirculation and filtration may be applicable to allergic manifestations connected with other antigenic dusts. The dust from feathers, from animal hair, and cotton produces definite sensitivity in some individuals and results in symptoms ranging from eczema to asthma. House dust is also known to produce definite allergic disturbances that are not referable to any of the known proteins. Recently one case of sensitivity to building material has been observed. It would seem entirely reasonable to suppose that, with efficient filtration of the air, the concentrations of these environmental dusts could be reduced so that there would be little danger of producing sensitivities in individuals not previously sensitized, and so that they would probably produce no allergic manifestations in those already sensitized.

Grateful acknowledgment for assistance in our work is made to the American Air Filter Company of Louisville, Ky., the Peoples' Gas Light and Coke Company of Chicago, and the General Electric Company.

IV. PHYSICAL FACTORS AFFECTING COMFORT

ALONZO P. KRATZ*

1. *Purpose of Air Conditioning.*—To a considerable extent the progress of civilization has been marked by man's success in his efforts to control his natural environment. When it is considered that human life can exist through only a small range of body temperature centered around 98.6 deg. F., it is not surprising that, insofar as the more vigorous climates are concerned, man's earliest efforts to control his environment were directed towards the conservation of bodily heat. The first attempts to accomplish this by artificial means represented the dawn of the era of air conditioning, for air conditioning in its broadest sense is not confined to summer cooling, but constitutes the complete control of atmospheric conditions in such a manner as to produce an environment most conducive to health and comfort during both winter and summer. The knowledge of the physiological effects of various air conditions on health is still incomplete. However, reasoning from the inference that bodily discomfort is not conducive to good health, it seems logical to conclude that the primary function of air conditioning is to provide a comfortable environment, on the assumption that such an environment is at the same time conducive to health.

2. *Development of Theories of Comfort.*—Most of the ideas regarding comfort were established in connection with the more restricted problem of heating, and were later developed to apply to both heating and cooling. In common with other arts and crafts, the practice of heating was firmly established and developed to a rather wide extent before any consideration was given to the underlying theory which was necessary for an orderly advancement in the field. The earlier theories were largely empirical, and furnished no means of explaining what constitutes comfort, or of establishing standards for evaluating or producing comfort. Since 1923, however, exhaustive studies at the Research Laboratory of the American Society of Heating and Ventilating Engineers, located at Pittsburgh, Pennsylvania, have resulted in the formulation of a theory of comfort that makes possible the establishment of standards, and affords a means of analyzing and designing both heating and cooling plants based directly on considerations relating to their effectiveness in producing comfort.

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3. *Conditions Required for Comfort.*—The modern theory of comfort is based on regarding the human body as a heat engine acting in such a manner that all bodily processes and functions ultimately result in the production of heat. The amount of heat thus generated varies with the amount of bodily activity engaged in. For an individual seated and at rest it is approximately 400 British thermal units (hereafter referred to as B.t.u.) per hour. With a moderate degree of activity, such as walking, the heat generated is increased to between 600 and 800 B.t.u. per hour, and for various degrees of heavier work it may easily exceed 1300 to 2000 B.t.u. per hour. For each degree of activity, however, this heat must be removed at the same rate at which it is generated, or else the normal functions of the body will be interfered with, and the results may range from some degree of discomfort to permanent injury or death.

The rate of heat loss is determined by a number of factors in the environment, such as temperature, relative humidity, and air motion. If the environment is not favorable to permit this loss of heat, the body attempts to compensate for it by making internal adjustments tending to accelerate or retard the loss. The most obvious of these adjustments are perspiration, shivering, and increases or decreases in skin temperature. As long as the individual is unconscious of such adjustments, a state of comfort exists, and discomfort begins as soon as the operation of the mechanism of adjustment becomes consciously apparent.

The proper function of a heating or cooling plant is, therefore, not to warm or cool the body, but to produce an environment in which the body is enabled to lose an amount of heat corresponding to that generated, without any conscious bodily adjustments having to be made.

4. *Factors Affecting Heat Losses from the Body.*—Heat is lost from the body by radiation, convection, and evaporation. Radiation has properties that are similar to those of light. It is transmitted through the atmosphere without appreciably heating the air through which it passes. The amount of heat lost by radiation is dependent on the body temperature and on the temperature of nearby surfaces, such as walls, furniture, and steam or hot water radiators. In the process of convection, cool air comes in contact with the body from which it absorbs heat by conduction. The air thus warmed rises, and is replaced by cooler air, which is in turn warmed. The amount of heat lost by convection, therefore, depends on the air temperature and the amount of air motion. Evaporation takes place from the skin and

respiratory tract, and is dependent on the temperature of the air and the amount of moisture, or the relative humidity of the air. Technically, what has been designated as moisture is not present in the form of water, but rather in the form of vapor.

Under winter conditions, in a normal atmosphere at 70 deg. F. and 30 per cent relative humidity, of the 400 B.t.u. per hour generated by a person seated and at rest, approximately 210 B.t.u. are lost by radiation, 90 by convection, and 100 by evaporation. These three factors, radiation, convection and evaporation, are subject to more or less independent control by varying the air temperature, air motion, and relative humidity; and by introducing warm or cold surfaces such as radiators and cold walls. However, no matter how these factors are varied, the sum of the separate heat losses must be approximately 400 B.t.u. per hour if comfort is to be maintained. The evaporation loss remains practically constant at 100 B.t.u. per hour. Hence, the control effected by the heating plant is accomplished by so adjusting the radiation and convection losses that the sum of the two is maintained at approximately 300 B.t.u. per hour.

Fundamentally, therefore, comfort may be regarded as depending on air temperature, air motion, and relative humidity, and the studies made at the Research Laboratory of the American Society of Heating and Ventilating Engineers have resulted in the establishment of a comfort chart that gives all of the various combinations of these factors which will be conducive to comfort. Briefly, with comparatively still air, the practical range for maximum comfort under winter conditions is from 70 deg. F. with 50 per cent relative humidity, to 73 deg. F. with 20 per cent relative humidity. Under summer conditions much wider variations can be tolerated, but a practical range for residence work is probably from 76 deg. F. with 70 per cent relative humidity to 82 deg. F. with 40 per cent relative humidity.

5. *Relation between Type of Heating System and Comfort.*—Warm-air furnace plants offer no hot surfaces in the rooms and hence the control of the heat loss from the body is accomplished almost entirely by controlling the convection. That is, if the individual feels too cold, indicating too great a heat loss, the air temperature is raised, thus reducing the loss by convection to the air. With this type of plant a temperature of 72 deg. F. with relative humidity of 30 per cent is probably the most practical and satisfactory. In the case of the conventional steam or hot water system, the radiators tend to offset part of the heat lost from the body by radiation; but the major part of the action is through the agency of convection, and in this

type of plant also the air temperature must be increased when the individual tends to feel too cold. Owing to the presence of the warm radiators it is probable that an air temperature somewhat lower than that required with the warm-air furnace plant will produce the same degree of comfort. Definite information on the amount of reduction permissible is lacking, however, and it is probable that it does not exceed 2 deg. F. The various types of convector heaters and concealed radiators produce effects lying between those of the warm-air furnace and the steam or hot water radiators.

A different type of system, known as the system of radiant heating is more popular in England than in the United States. In this type of plant, low temperature panel radiators are employed. These panels cover a large portion of the walls and ceiling of the room. Very little convection occurs, and the major part of the action is through offsetting the radiant heat loss from the body. In this case, air temperatures as low as 55 deg. F. may be sufficient for comfort, and comfort is attained by raising the mean radiant temperature, or, in other words, the temperature of the panel surfaces. The comfort chart is not applicable where this type of heating is used.

6. *Effect of Cold Walls and Windows.*—As previously mentioned, the bodily heat loss of 300 B.t.u. per hour by combined radiation and convection is subdivided into 210 B.t.u. per hour by radiation and 90 B.t.u. per hour by convection. Hence it is evident that radiation is the more important of the two, and the temperature of the surrounding walls and objects may have a more important bearing on the comfort of the individual than is usually recognized. The temperatures that may exist at the inside surface of a typical exposed frame wall, consisting of lath and plaster, 3⅝ in. studding, sheathing, and clapboards, are shown in Fig. 1. Here, when the outdoor temperature was 8 deg. F. below zero, shown as an indoor–outdoor temperature difference of 80 deg. F., the temperature of the inside surface of the exposed wall was 60 deg. F. with no wind. With a wind having a velocity of 10 m.p.h. blowing on the exposed surface, the temperature of the inside surface was only 52 deg. F. Under these conditions a slight modification must be made in the interpretation of the comfort chart. With three exposed walls, and the room air at 72 deg. F., if the inside wall surfaces are at 52 deg. F. the air temperature must be increased to 78 deg. F. in order to obtain the same degree of comfort that would exist with both air and inside wall surfaces at 72 deg. F.

Naturally, the effectiveness of the wall as a heat insulator is reflected in the temperature of the inside surface. Poorly insulated

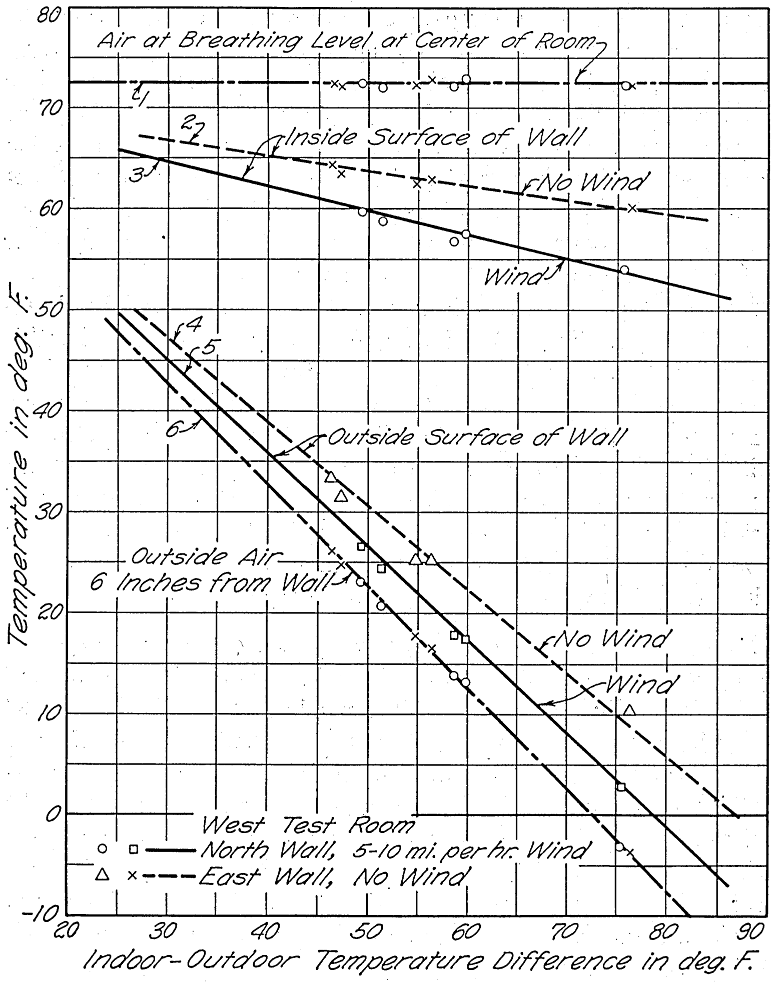

FIG. 1. INSIDE AND OUTSIDE WALL SURFACE TEMPERATURES

walls result in low surface temperatures. The typical frame wall used for Fig. 1 had an overall heat transmission coefficient of approximately 0.26 B.t.u. per sq. ft. per hour per deg. F. If this wall had been insulated with some type of filled insulation, so that the overall heat transmission coefficient was reduced from a value of 0.26 to one of 0.064, the inside surface temperature with a 15-mile per hour wind and with 8 deg. F. below zero outdoors would have been 68 deg. F. instead of 52 deg. F. In this case a room air temperature of 73 deg. F. would

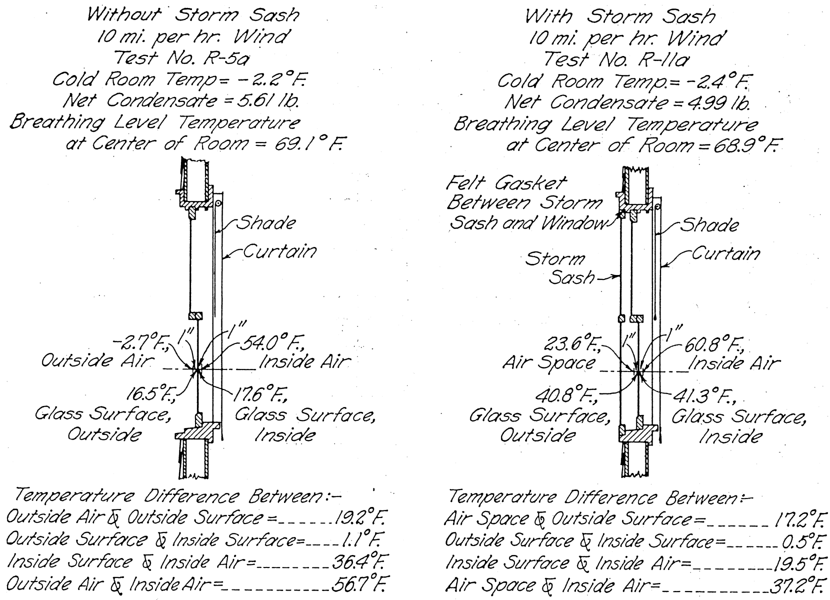

FIG. 2. TEMPERATURE GRADIENTS THROUGH GLASS IN WINDOWS

have produced the same degree of comfort as that obtained with both air and wall surfaces at 72 deg. F.

A very common source of discomfort arises from the presence of cold glass surfaces. Figure 2 illustrates the inside glass surface temperatures obtained with two windows, one equipped with tightly-fitting storm sash, and the other without the storm sash. In the latter case, with an outdoor temperature 2.7 deg. F. below zero the temperature of the inside surface of the glass was 17.6 deg. F. Under the same conditions in the former case the temperature of the inside surface of the inside pane of glass was 41.3 deg. F. Hence, it is evident that a few windows may be more detrimental to comfort than considerable areas of exposed wall, and that the addition of storm sash may materially increase the comfort in a room having a number of windows.

Two rather important conclusions may be drawn from the discussion in the preceding paragraphs. First, if a considerable area of cold walls and windows is present, the reading of a thermometer, ostensibly giving the temperature of the air in the room, may not afford a reliable criterion for judging the comfort of the occupants. Higher air temperatures are demanded to offset cold exposures, and,

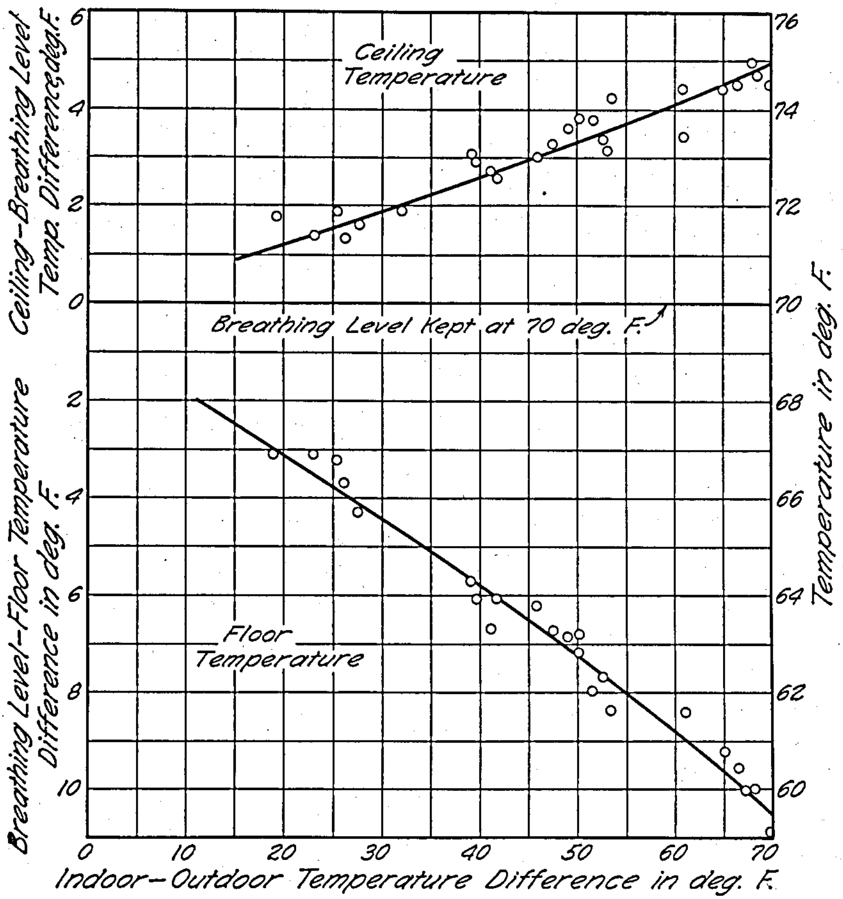

FIG. 3. CURVES OF FLOOR, BREATHING LEVEL, AND CEILING TEMPERATURES FOR LIVING ROOM IN RESEARCH RESIDENCE, SECOND INSTALLATION

in addition, a glass thermometer is less affected by radiation than is the human body. Hence a thermometer is less influenced by the presence of cold walls. Second, adequate insulation and tightly-fitting storm sash, besides directly saving heat loss from the structure, have an important function in increasing inside surface temperatures, and thus directly in improving comfort conditions. Furthermore, by permitting the use of lower air temperatures to attain the same degree of comfort, the use of insulation and storm sash may result in an additional indirect saving of heat.

While the preceding discussion has been made from the standpoint

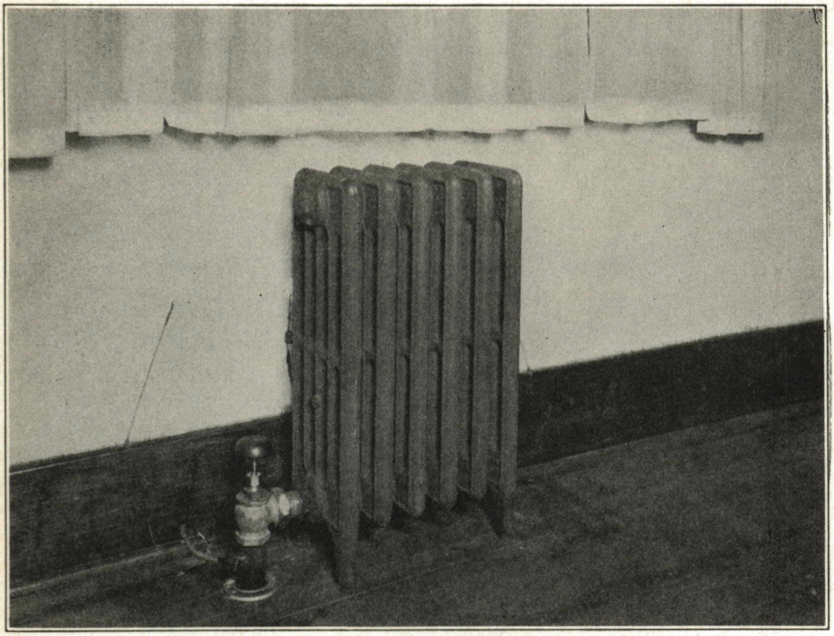

Fig. 4. Six-section, 26-in. Height, 5-Tube Column Radiator in Cold Room Testing Plant

of heating, it is equally applicable to the case of cooling. The presence of hot walls and ceilings makes necessary the use of lower air temperatures to attain the same degree of comfort, thus imposing more load on the cooling plant. In addition to this, the mere presence of the hot walls and ceilings indicates excessive heat gain by the structure, and therefore additional load on the plant.

7. *Effect of Temperature Gradients in Rooms.*—In addition to the effect of cold walls, variations in the temperature of the air at different levels in the room may still further operate to discount the value of the reading of a thermometer as a reliable index of comfort. The most common location for such a thermometer is on an inside wall at a height of about 5 ft. from the floor. If the temperature of the air is not uniform from the floor to the ceiling, the warmest air will almost invariably be found near the ceiling. Hence, the reading of the thermometer may indicate a temperature sufficiently high for comfort while the greater part of the body may be surrounded by air at a temperature too low for comfort. This is particularly true for individuals seated.

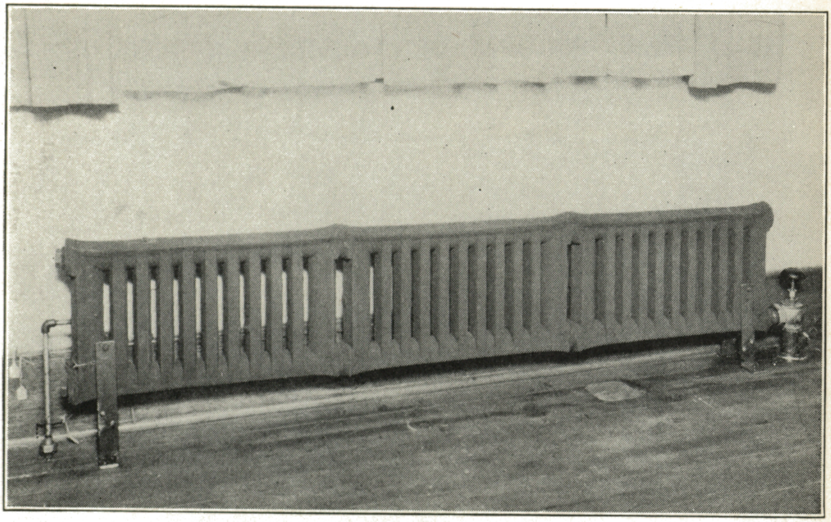

Fig. 5. Three-section, 21-Sq. Ft., Wall Type Radiator Located Near the Floor in Cold Room Testing Plant

The condition of non-uniform temperatures at the different levels in a room is not confined to one type of heating plant alone, but is more or less common to all types. Figure 3 shows results obtained from a well-designed gravity warm-air system in which the number of air recirculations varied from 1.8 in mild weather to 3.0 in zero weather.

It may be noted that with zero weather outdoors, or with an indoor–outdoor temperature difference of 70 deg. F., the thermometer at the 5-ft. level indicated a comfortable temperature of 70 deg. F. while the temperature of the air near the floor was approximately 60 deg. F. Hence, the body of a person standing would be surrounded by air at a mean temperature of 65 deg. F., and of a person sitting, at a mean temperature somewhat less than 65 deg. F. In this case the room was not definitely uncomfortable for some people, but was so for others.

By increasing the temperature to 73 at the 5-ft. level the room could be made definitely comfortable. This may be regarded as a border-line case in which the inside surfaces of the exposed walls were at approximately 60 deg. F. It may also be noted that in milder weather the temperature difference between the floor and the 5-ft. level decreased rather rapidly.

Two characteristic types of radiators are shown in Figs. 4 and 5. Figure 4 represents a 5-tube radiator, 26-in. high, rated at 21 sq.

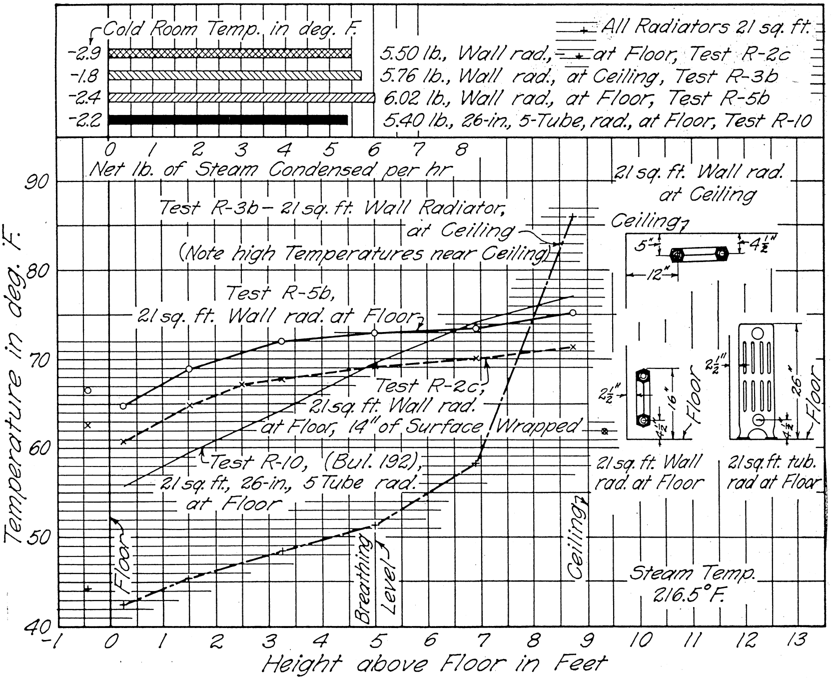

FIG. 6. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR UNENCLOSED RADIATORS IN COLD ROOM TESTING PLANT

ft. of radiation; Fig. 5 represents a wall type of radiator, also rated at 21 sq. ft.

The temperature variations from the floor to the ceiling, and the steam condensations, obtained with the tubular radiator located in the usual position under a window, and with the wall radiator located in two positions, are shown in Fig. 6. The tubular radiator produced a temperature of only 55 deg. F. at the floor with 70 deg. F. at the 5-ft. level, and 77 deg. F. at the ceiling. Hence the living zone, or the portion of the room below the 5-ft. level, was uncomfortably cold, although the thermometer at the 5-ft. level indicated comfort.

Using the wall type of radiator with the same rated heating surface in a similar position under the window the room was heated to 73 deg. F. at the 5-ft. level and 65 deg. F. at the floor. When the surface of this radiator was decreased so that the temperature at the 5-ft. level was 70 deg. F., comparable with that obtained with the tubular radiator, the temperature near the floor was 60 deg. F. as compared with the 55 deg. F. obtained with the tubular radiator. The tempera-

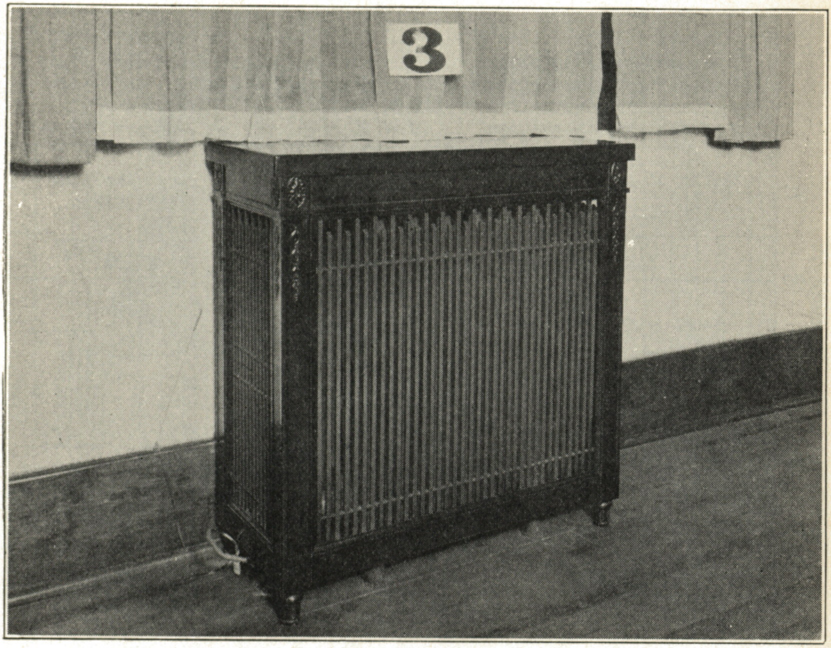

FIG. 7. RADIATOR ENCLOSURE NO. 3 TESTED IN COLD ROOM TESTING PLANT

ture in the whole of the living zone was higher than that produced by the tubular radiator, while the steam condensation was practically the same. Hence, it is evident that the temperature distribution in a room is greatly influenced by the type and location of the heater, and that a long, low, and flat radiator is more conducive to comfort than a short, high, and comparatively thick one.

As a further illustration of an extreme case, Fig. 6 shows also the temperature gradient obtained with the wall radiator located horizontally just below the ceiling. In this case the temperature near the floor was 42 deg. F., while that at the 5-ft. level was only 51 deg. F., and 3 in. below the ceiling it was 86 deg. F. Within ¼ in. of the ceiling a temperature of 103 deg. F. was indicated. From the standpoint of satisfactory heating these conditions are intolerable.

Many means have been adopted to improve both the appearance and the performance of radiators by adding various types of shields, covers, and enclosures. Two of the latter will be considered here as being more or less characteristic. Figure 7 shows one of the more open types of enclosures, and Fig. 8 the room temperature gradient obtained when this enclosure was used in connection with the tubular radiator

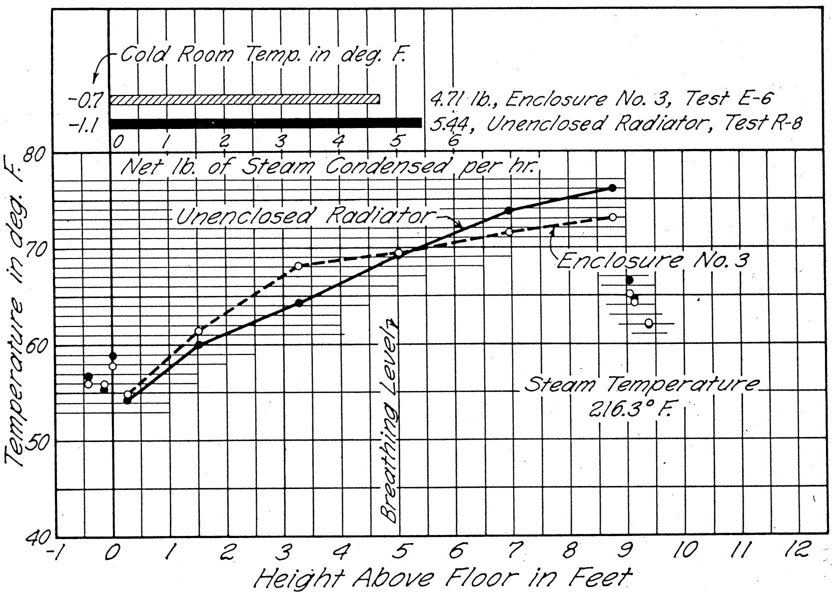

FIG. 8. ROOM TEMPERATURE GRADIENT AND STEAM CONDENSING RATE FOR RADIATOR ENCLOSURE NO. 3

shown in Fig. 4. The temperature gradient for the unenclosed radiator is also given for the purpose of comparison.

In this case the use of the enclosure resulted in a higher temperature, and hence more comfortable conditions in the living zone, and in lower temperatures at the ceiling than those obtained with the unenclosed radiator. Furthermore, these results were obtained with less steam condensation in the case of the enclosed radiator. Thus this type of enclosure proved to be both effective and economical.

Figure 9 shows a type of enclosure with restricted air inlet and outlet, and Fig. 10 the results obtained with this enclosure. In this case, while the enclosed radiator showed less steam condensation than the unenclosed, no improvement in the temperature gradient was obtained, and the enclosed radiator failed to raise the temperature at the 5-ft. level to the 70 deg. F. obtained with the unenclosed radiator. The temperature at the 5-ft. level could have been increased by increasing the size of the radiator, but no improvement in the temperature gradient would have resulted. Hence, this type of enclosure can not be regarded as either effective or economical.

From the preceding discussion it may be concluded that a thermometer placed at the 5-ft. level may give an incorrect indication of

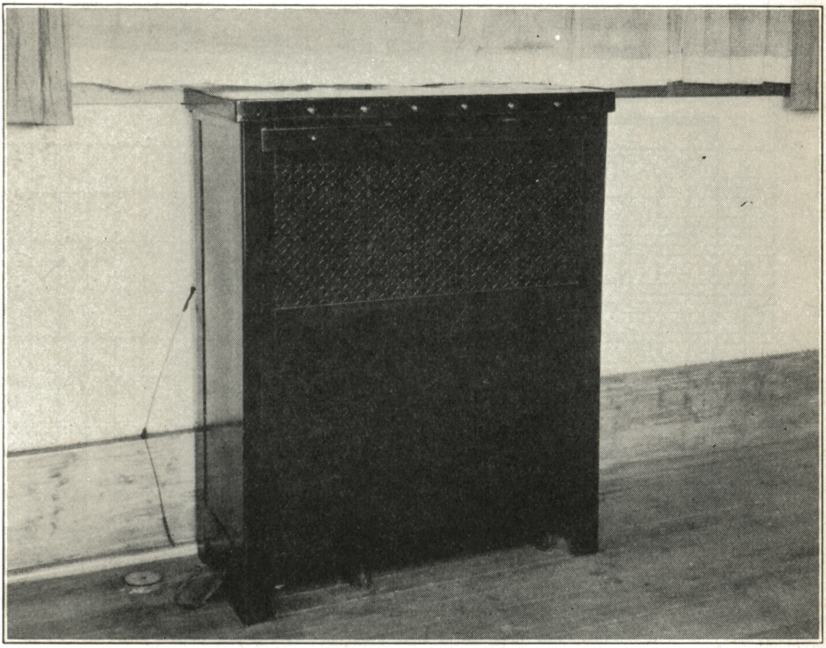

FIG. 9. RADIATOR ENCLOSURE NO. 10 TESTED IN COLD ROOM TESTING PLANT

the degree of comfort experienced in the room, and that while a certain amount of deviation from the true index is characteristic of nearly all heating plants, the extent of the deviation is influenced by the type of the heating units or heating system. All indications point toward the fact that it would be better practice to locate the thermometer at the 30-in. level rather than at the 5-ft. level.

The same principles apply even more forcibly to the location of the control thermostat than to the location of the thermometer. If the thermostat is placed at the 5-ft. level, and conditions are not complicated by other factors, a comfortable temperature may be maintained at the 5-ft. level, but a lower temperature will exist in the living zone resulting from the normal temperature gradient in the room. Unfortunately this situation is further complicated by other factors.

The thermostate cannot act to open the dampers, or start the burner, until the air at the 5-ft. level has cooled to the temperature predetermined by the setting of the thermostat. Meanwhile the dampers remain closed, or the burner remains off, and practically no heat enters the room. At the same time cold outdoor air enters

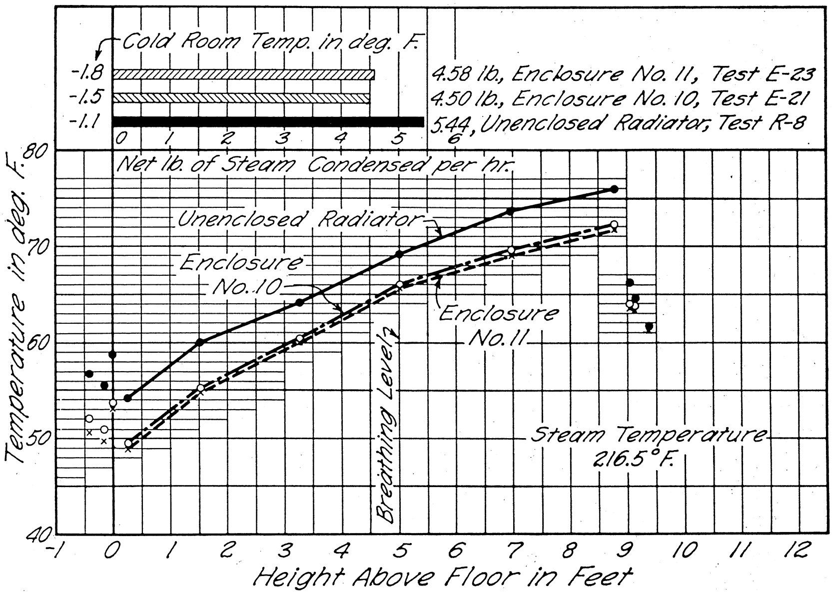

FIG. 10. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR RADIATOR WITH ENCLOSURES NOS. 10 AND 11

through cracks in the window and door frames, and the air in contact with the cold glass becomes chilled. All of this cold air collects as a pool near the floor, and, if the off-periods extend over a considerable time, the temperature gradient becomes much more pronounced than it is during the periods when the heater is in actual operation. Hence, during the off-periods the room may become decidedly cold before the thermostat acts to furnish more heat.

This condition, sometimes known as the "cold 70," is more prevalent with gas- and oil-fired systems than with coal-fired, because in the latter case the heat stored in the fuel bed continues to be supplied to the room even when the dampers are closed, but in the former case the heat supply ceases soon after the burner becomes inoperative. In the case of forced-air heating systems this condition can often be improved by placing the thermostat at the 30-in. level, and adjusting it to decrease the length of the off-periods. In the case of gravity warm-air systems and steam and hot-water systems, it is not so certain that this procedure will entirely correct the difficulty.

8. *Humidification.*—Medical opinion concerning the physiological effects of atmospheres of low relative humidity is not entirely

unanimous, and some authorities claim that for normal individuals any direct relation between relative humidity and health has not been definitely proved. Hence it appears that comfort affords the best argument in favor of adequate humidification. At any rate, comfort seems to offer the most tangible standards of measurement. The comfort chart previously mentioned indicates that maximum comfort will be attained with a temperature of 70 deg. F. and relative humidity of 50 per cent. If the relative humidity is decreased to 20 per cent the same degree of comfort will be maintained if the air temperature is increased to 73 deg. F. Relative humidity below 20 per cent will produce a sensation of dryness, but there seems to be evidence to support the belief that it may be as low as 20 per cent without undue effect on comfort, although it is customary to recommend 40 per cent as being the most desirable value.

No saving in fuel resulting from increasing the relative humidity can be demonstrated. While it is true that an increase in relative humidity permits the use of somewhat lower air temperature to maintain the same degree of comfort, thus representing a potential saving in fuel, the additional heat required to evaporate the water almost exactly offsets the saving effected by the decrease in air temperature. Hence, the importance of humidification has been to a certain extent overrated.

Since the recommendation of 40 per cent relative humidity is usually regarded as conservative, the implications involved in the maintenance of this amount may well be examined. The amount of infiltration of outdoor air into the average well-built house without weatherstripped windows amounts to approximately one air change per hour. Figure 11 shows the weight of water that must be evaporated in order to maintain various percentages of relative humidity in 10 000 cu. ft. of space for different outdoor temperatures. From these curves it may be observed that it requires 11.5 gal. of water per 24 hr., in order to maintain 40 per cent relative humidity in 10 000 cu. ft. of space when the outdoor temperature is zero. A medium-sized residence will contain approximately 15 000 cu. ft. of heated space, and would therefore require evaporation of the order of 17 gal. per 24 hr. If weatherstripped windows or tightly-fitting storm sash are used the air infiltration may be materially reduced, and the evaporation requirements may be correspondingly reduced to 8 gal. per 24 hr. or less.

The usual warm-air furnace pan will evaporate about 2 gal. per 24 hr., and under the best conditions will not evaporate more than 5

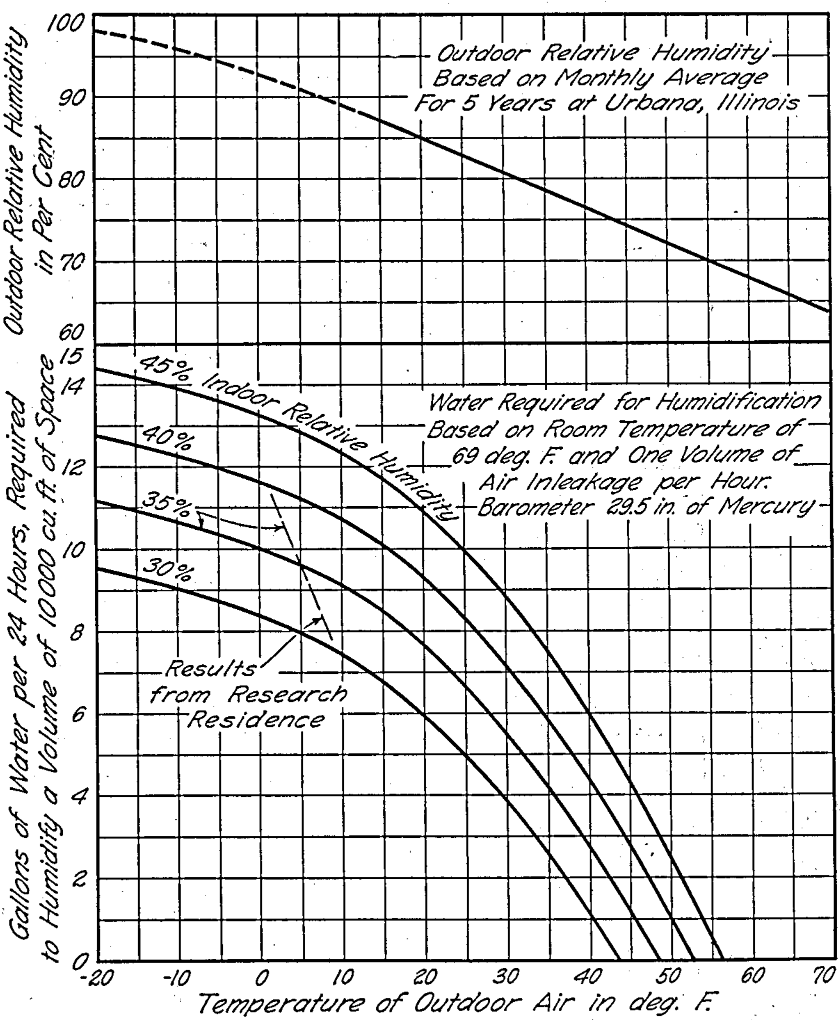

FIG. 11. CURVES FOR EVAPORATIVE REQUIREMENTS FOR HUMIDIFIERS

gal. per 24 hr. A pan placed on the dome of the furnace will, under the most favorable conditions, evaporate as much as 14 gal. per 24 hr. Open pans used in connection with steam radiators will evaporate approximately 0.75 gal. per sq. ft. of water surface per 24 hr. The evaporation of 17 gal. per 24 hr. would therefore require 23 sq. ft. of surface, or, since the average pan is about 10 in. wide, it would require the equivalent of a pan 27 ft. long. Hence, it is evident that

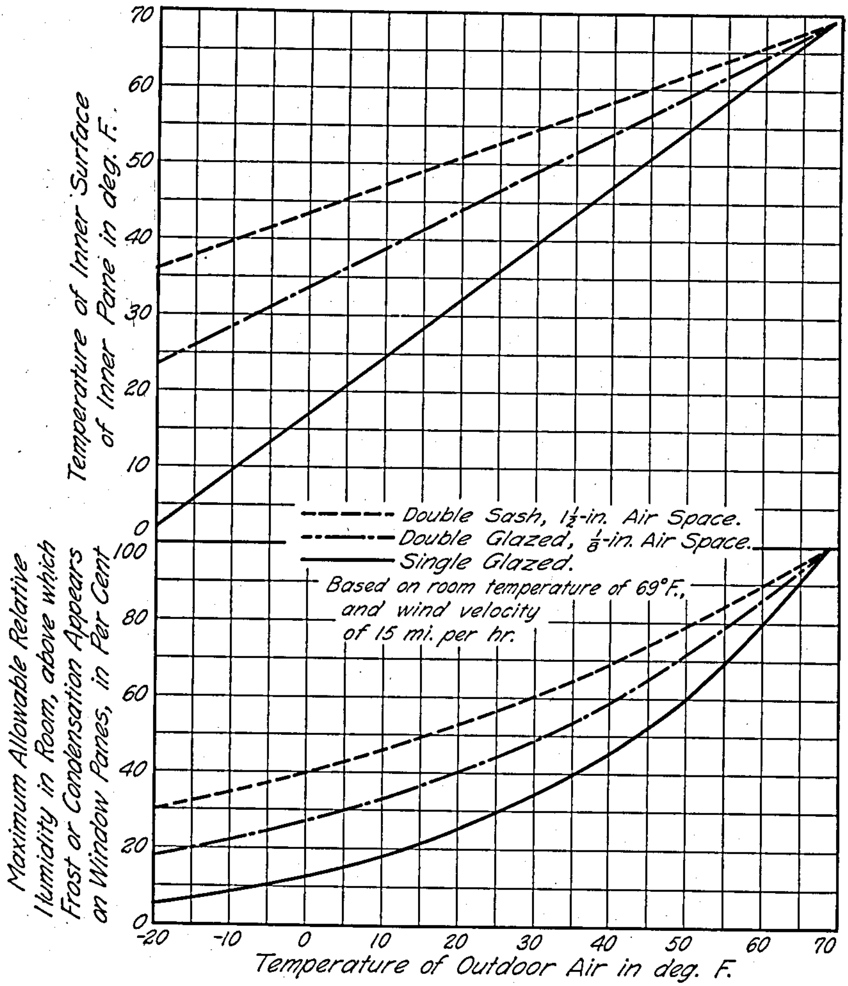

FIG. 12. CURVES FOR GLASS SURFACE TEMPERATURES AND LIMITING VALUES FOR RELATIVE HUMIDITY

40 per cent relative humidity cannot be maintained in zero weather without some special provision in the design of the heating system or the use of auxiliary apparatus.

If water vapor is present in the air, condensation will occur on any surfaces that have a temperature less than the dew point. Figure 12 shows the percentage relative humidity at which condensation will appear on three types of windows for different outdoor temperatures. From these curves, if 40 per cent relative humidity is maintained indoors, condensation will start to appear on windows composed of a single thickness of glass when the outdoor temperature drops to 35 deg. F.

If tightly-fitting storm sash are used the outdoor temperature can drop to zero before condensation starts. Stated differently, with zero weather outdoors, indoor relative humidity of 40 per cent can be maintained without material condensation if storm sash are used, but not more than 15 per cent relative humidity if storm sash are not used.

Where single thickness windows are used without weather-stripping a protective layer of cold air enters through the cracks around the sash that tends to dilute the more humid indoor air at the surface of contact. Hence, it is sometimes observed that less condensation occurs on windows that are not weatherstripped than on corresponding windows that are weatherstripped.

Poorly insulated walls may have surface temperatures sufficiently low to result in condensation when relative humidity as high as 40 per cent is maintained. Even with fairly well insulated walls, if the temperature in the house is allowed to drop materially at night, the temperature of the inside surfaces may decrease to such a point that, if the furnace is forced the next day, enough evaporation may take place in a short period to raise the relative humidity high enough for condensation to occur on the exposed walls. In the case of frame walls, it is practically impossible to have the window frames, baseboards, etc., so tight that no air leakage occurs from the room into the studding spaces. Condensation may then take place on the inside surface of the sheathing, and water may ultimately soak through to the clapboards or stucco.

Since considerable difficulty may be experienced with high relative humidity, it hardly seems practical to recommend 40 per cent for all residences irrespective of their particular features of construction. The lowest boundary of the comfort chart, 30 per cent, is safely within the limits of comfort, and there is reason to believe that within the proper temperature limits 20 per cent is not noticeably too dry for comfort. Hence, for residences with average construction and not equipped with storm sash, 25 per cent relative humidity in zero weather would seem to be a reasonable value to provide comfort and to avoid some of the difficulties liable to result from attempts to maintain unduly high humidity.

9. *Importance of Good Construction.*—While the theory that the attainment of comfort results from creating an environment in which

the heat loss from the body can offset the heat generated, without the necessity for conscious bodily adjustments, is simple in its conception, it is also evident that a considerable number of factors influence the attainment of this end. Some of these factors are inherent in the design and operation of the heating or cooling plant. Most of them, however, are influenced directly by the character of the structure in which the plant is installed. Thus features of good construction like air-tight walls, heat insulation, weatherstripped windows and storm sash all have significant bearing on comfort and the ability to maintain proper humidification, as well as on the economical operation of the heating plant.

Considerable effort has been directed toward the design of the heating or cooling plant, and, until recently, not enough attention has been paid to the structure housing this plant. In a properly-constructed building the problem of the heating or cooling plant is comparatively simple, and it should be the function of the air conditioning engineer to arouse the public interest in good construction and to insist on such good construction as being inseparable from the proper functioning of the heating or cooling plant itself.

V. AIR FILTERS IN AIR CONDITIONING SYSTEMS

FRANK B. ROWLEY*

The purpose of this paper is primarily to discuss air filters and their application to air conditioning systems, but in order to get the proper understanding of the air filter problem it is desirable to know something about those materials which the filter has to handle. Impurities in the air may be either in the form of gases or solids, and while filters might be devised for absorbing the impure gases in the air, those which will be considered here are for the purpose of removing the solid matter, usually in the form of dust.

No air is free from dust. It comes from practically every process which will cause the abrasion or disintegration of materials, and is thrown into the air by mechanical agitation, air movement, or any method which will tend to disturb the dust particles. It is created and stirred up by moving vehicles, and a great many industrial processes, and a constant source of atmospheric contamination is the smoke from chimneys from either industrial plants or private homes. Once the dust particles are in the air they will remain in suspension for periods of time varying from a few seconds up to several years depending upon the size and density of the particles. Dust particles range in size from those which can be easily seen with the naked eye down to particles which are so fine that they are only visible by the ultra-microscope. As a practical means of rating the dustiness of the air the average size of the particles and the concentration per cubic foot is often used.

The diameter of dust particles is usually given in microns. As a note of explanation, a micron is one one-thousandth of a millimeter, or one twenty-five-thousandth of an inch. The average human hair is about 60 microns in diameter and a particle 10 microns in diameter is the smallest that is visible to the naked eye. The dust particles in which we are most interested are from perhaps 100 microns down to ½ micron.

The concentration of dust particles in the air varies over wide ranges, depending upon surrounding conditions. In clean country air, and especially after it has been washed by rain, the concentration may be only a few thousand particles per cubic foot. In average outside air, if there is such a condition, it will probably range around 500 000 particles per cubic foot, and in many city areas it will run as high as 4 to 5 million particles per cubic foot. It is not uncommon to find air

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in dusty streets or around a smoky city which will run above 5 million particles per cubic foot, and in some surveys over 20 million particles have been common. The dust level in many industries, and especially in the dusty operations of those industries, will run into many million particles per cubic foot.

The fact that there is dust in all air need cause no great concern, as moderate concentrations of many dusts which are common in the atmosphere actually serve a useful purpose. It is the reflection of sunlight by the dust particles in the air which gives the morning and evening twilight period and the colorful sunrise and sunset. Dust particles also form the nuclei for rain drops, and if there were none to start the initial condensation of moisture the air would hold something like seven times as much moisture as it now holds before condensation starts. Under this condition we would probably be surrounded by a heavy, hazy supersaturated atmosphere and any initial precipitation would be followed by terrific rains.

While we are not concerned about the normal concentration of dust in reasonably pure air, there are objections to certain kinds of dust and to excessive concentration of any kind of dust. These objections may be from an economic or health standpoint. The economic loss may be due to the actual value of the dust itself in the air, in which case the recovery may be a necessary part of the industrial process, or the loss may be due to the effect on surrounding property and vegetation. The effect on the exterior of buildings and other property in dusty industrial sections is a common sight. Likewise, most people are familiar with the excess dry cleaning and laundry bills caused by smoke and dirt in outside air, as well as in the interior of buildings.

Dust may have either a direct or an indirect effect on health. The dust and smoke over any given area shuts off the sunlight and is particularly effective in reducing ultra-violet rays. To aggravate the condition the dust particles form the nuclei for fine drops of moisture which cause the heavy fog over some industrial areas. Often these fine drops of moisture in the fog become coated on their exterior with the oily soot from the products of incomplete combustion, and in this condition re-evaporation is retarded, thus aggrevating the fog nuisance.

Any direct effect which dust may have on health depends upon its chemical composition, the nature and size of its particles, the concentration in the air, and the length of exposure to dust and conditions under which it is breathed. Some forms of dust such as lead, arsenic, mercury, etc. are somewhat soluble in the blood stream, and act as poisons to the system. Impurities of this nature may affect the health not only by being taken into the lungs but by any contact from which they may be absorbed into the blood stream. In order that a dust may be harmful to the lungs or respiratory tract it is necessary, first, that it be of such size that it enters the lungs, and second, that it be either poisonous or of such physical properties that it will be irritating. The respiratory tract leading to the lungs contains very efficient air-filtering media. The large particles are first screened out as the air enters the nose, and the screening process is progressive until only those particles of 6 or less microns in diameter actually enter the lungs. The damage which such particles may do depends entirely upon the nature of the dust. In most cases the organic dusts are less harmful than the inorganic dusts, and it is usually considered that those dusts which are more nearly assimilated to the composition of the human body are the least harmful. Dust particles from 14 to 16 microns in diameter are screened out in the upper nasal passages, and some of them may cause an irritating effect known as hay fever. Pollen from various plants which are known to cause hay fever are really dust deposits in the air, and may be treated as such.

The dust problem is one which is always present for the air conditioning engineer and often requires a great deal of thought and planning to meet it successfully. There are at least three general solutions to the problem. The first and most effective method is to prevent the dust at its source. Unfortunately, however, this is usually beyond the scope of a particular air conditioning job, and something which must come about by a slow process of public education and the enforcement of specific regulations on the particular offenders. The air conditioning engineer must usually take the surrounding conditions as he finds them and give the air proper treatment for his requirements. The second possibility is to select the air from as clean a source as possible and thus reduce the cleaning problem. In most cases the dust concentration is different for various sides of the building and at various elevations. By some care in selecting the source of supply much may be gained in air purity. The third and final possibility is to clean the dirt and dust out of the air as it goes through the air conditioning system. This is primarily the function of air filters, and the remainder of this paper will be devoted to a discussion of the types and characteristics of such devices.

There are many possible ways of cleaning dust out of air, but the filters which are in common use may usually be classified as dry filters or viscous coated filters. In this classification the dry filter implies a type that actually cleans the dust out of the air due to the fineness of its mesh. Thus a chemical filter paper or a fine felt pad may be so closely woven that the air will pass through, but the dust will be retained in the mesh. In the viscous type of filter some of the dust might actually be screened out but the filtering action is one of impinging the air on to a viscous coated surface which will retain the dust particles. These surfaces may be built up in many ways, but the retention of the dust is due to the fact that it sticks to the surface, and not to the fact that the particles cannot pass through the openings. In either class of filter there are certain fundamental requirements if it is to be successfully used in an air conditioning system. Four of these requirements are:

(1) High efficiency, or dust arresting power.
(2) Low resistance to the flow of air through the filter.
(3) High dust-holding capacity, or long life in service.
(4) Economy in first cost and upkeep.

In addition to these there are other requirements such as freedom from odors, fire resistance, uniform efficiency over a wide range of air temperatures, low moisture absorption, etc.

The results which may be expected from any type of filter will naturally vary over a wide range, depending upon the specific properties of the filter and the operating conditions. In order to give comparative ratings the American Society of Heating and Ventilating Engineers has adopted a standard test procedure. A uniform dust mixture is selected consisting of 50 per cent by weight of carbon black and 50 per cent of Pocahontas ash passed through a 200 mesh screen. This dust is mixed into the air stream leading to the filter at a uniform rate, the arrestance and pressure drop across the filter being recorded throughout the test. The test results are plotted on coördinate paper with time and rate of dust feed as abscissas and arrestance and pressure difference across the filter as ordinates. Tests by this method are accelerated tests with a specific type of dust which may not in all cases correspond to operating conditions, but the results do show a comparison between different filters, and in general show the characteristics of individual types.

Figures 1 to 7 show the results for a series of filters tested by this method. The filters of Figs. 1 and 2 are strictly of the dry type, in which the mesh is sufficiently fine to screen out reasonable proportions of the dust. Filter 3 is of a dry fibrous material of a coarse mesh, such that the dust could easily pass between the fibers. Filters 4, 5, and 6 are of a viscous-coated fibrous material, in which the dust is

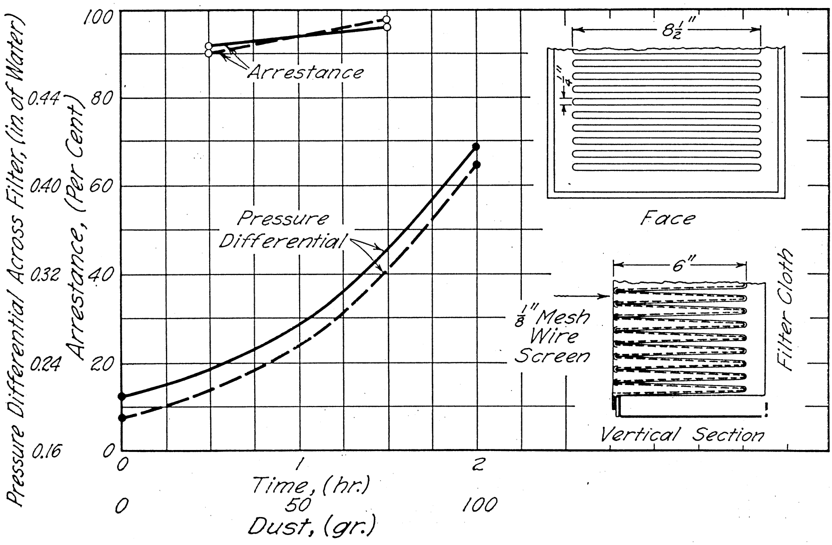

Fig. 1. Results With Felt Cloth Dry Filter

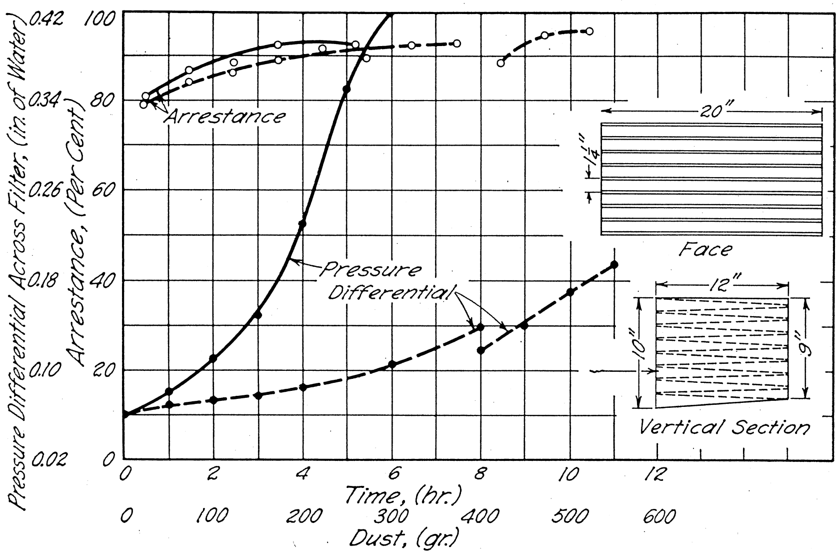

Fig. 2. Results With Cotton Batting Dry Filter

taken out by adhering to the fibers of the filter. Filter 7 is of a cellular type in which the walls of the cells are coated with a viscous material and the dust is removed by impingement against these walls. The results of these tests show typical characteristics which may be expected of each type.

Referring to the filters used in Test 1, the filtering material consisted of a rather close mesh felt cloth approximately $\frac{1}{16}$ of an inch thick, and weighing 7 ounces per square yard. The filter cloth was placed in a filter frame 6 inches deep, and plaited as shown in the accompanying figure. The spacing between the plaits was such that the actual exposed area of the filter cloth was twenty times the cross sectional area of the filter duct, thus giving an air velocity through the filter proper of $12\frac{1}{2}$ feet per minute. Two filters were tested, and the results shown by the curves are typical for filters of this type. The dust arrestance is very good. The air pressure drop across the filter is high, and the dust-holding capacity is low. The efficiency increased during the test due to the fact that the pores or openings between the fibers gradually became smaller, and thus stopped more of the dust. This reduction in the size of the openings is also indicated by the rapid rise in resistance to air flow through the filter. A filter of this type would be practical where a high efficiency is desired, where the total amount of dust to be handled is small, and where a high pressure drop may be allowed across the filter. As will be shown later a filter of this type will in general give increased efficiencies and reduced pressure drop through the filter as the air velocities are reduced.

The filter from which the data for Fig. 2 was obtained is a dry type of filter built up of a layer of cotton batting approximately $\frac{1}{4}$ inch thick. This cotton material was of a loose fiber on one side, but the other side was more densely matted and glazed. The material was plaited in a frame 12 inches deep, with the plaits approximately $1\frac{1}{4}$ inches apart, thus giving an actual filter area of 19.2 times as much as the cross sectional face area of the filter. Thus the actual velocity of air through the filter was 13 feet per minute. This filter was tested with the air passing through in each direction. The solid line curves represent the results obtained when the air entered the filter media from the glazed side and the broken line curve represents the results obtained when the air entered from the loose matted side. The break in the broken line was due to the fact that the filter stood over night after the first 8 hours of test, and the results for the first hour in the morning did not check with the extended curve for the previous results. This was found to be true in the case of many viscous filters, and makes it necessary, in so far as direct comparisons are concerned, to run a filter rating test either continuously or else at uniform and equal intervals. The interesting comparison between the two tests on this filter is found in the rapid rise in pressure drop across the filter for the test in which the air enters from the glazed side of the cotton as compared with the pressure drop for the test when it enters from the opposite side. This is due to the fact that the fine mesh of the fibers on the glazed side was most effective in screening out the dust particles, and when the air entered from this side the fine openings soon loaded up with dust and raised the pressure differential. When the air entered through the loosely-packed fibers a part of the dust was taken out by these fibers, and the capacity of the fine mesh fibers was reserved for the final filtering of the air. This principle of progressively increasing the density of fiber pack as the air passes through the filter is commonly used in the viscous-coated type. The differences in efficiencies between the two tests on this filter are not great, but the performance is very much better when passing the air through in the proper direction. In either test the arrestance is less than for the felt material of filter No. 1, but the dust-holding capacity is very much greater, and for the second test the pressure drop at the end of 11 hours was approximately the same as that at the beginning of the test of Fig. 1.

The filter used for Test No. 3 was built of metal wool loosely packed in the dry form. This material was coarse, loosely packed, and furthermore it was used as one flat sheet perpendicular to the air stream and not corrugated or plaited as filters Nos. 1 and 2, thus giving an air velocity through the filter of 250 feet per minute. The combination of coarse, loosely packed dry fibers and high air velocities make such an arrangement practically useless as an air filter. The results, as shown in the curve, indicate a low pressure drop and an impractically low arrestance. The results are valuable only in that they show how filters should not be designed.

Figures 4, 5, and 6 show the results from tests of viscous-coated fibrous filters. They are all approximately 2 inches thick with fiber sizes and density of fiber pack as given on the data sheet. A comparison of test results for these filters shows that for filter No. 4, which had the most densely packed and finest fibers, the arrestance was higher and the air pressure drop was greater across the filter than for filters Nos. 5 and 6, for which the fibers were coarser and more loosely packed. The rise in resistance throughout the test for No. 4 was also greater than for either of the others. In filter No. 6

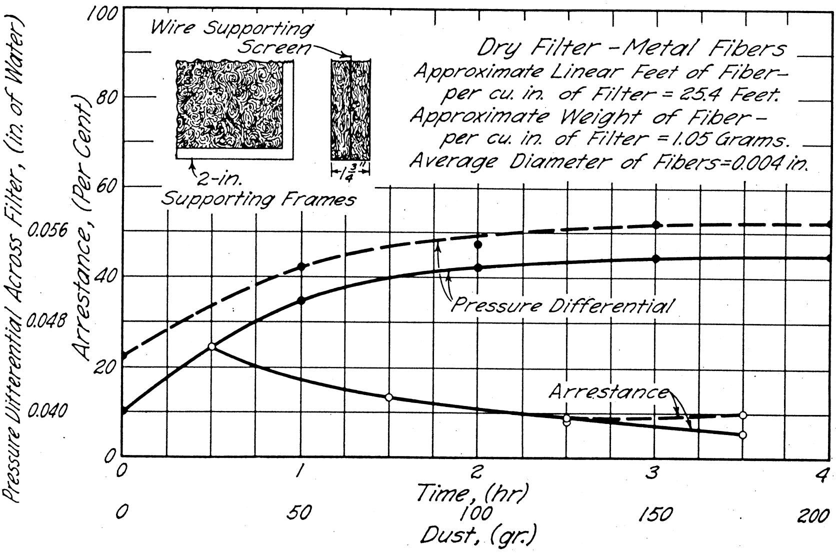

FIG. 3. RESULTS WITH METAL WOOL DRY FILTER

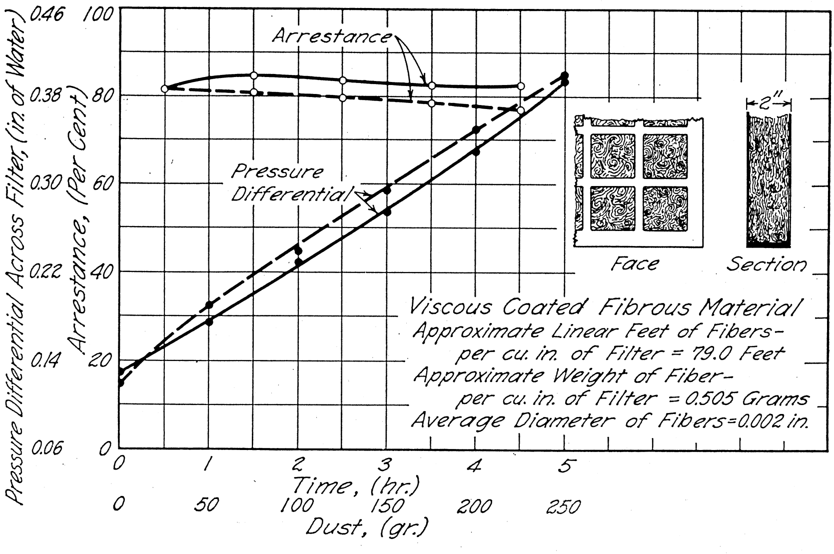

FIG. 4. RESULTS WITH VISCOUS COATED FIBROUS FILTER

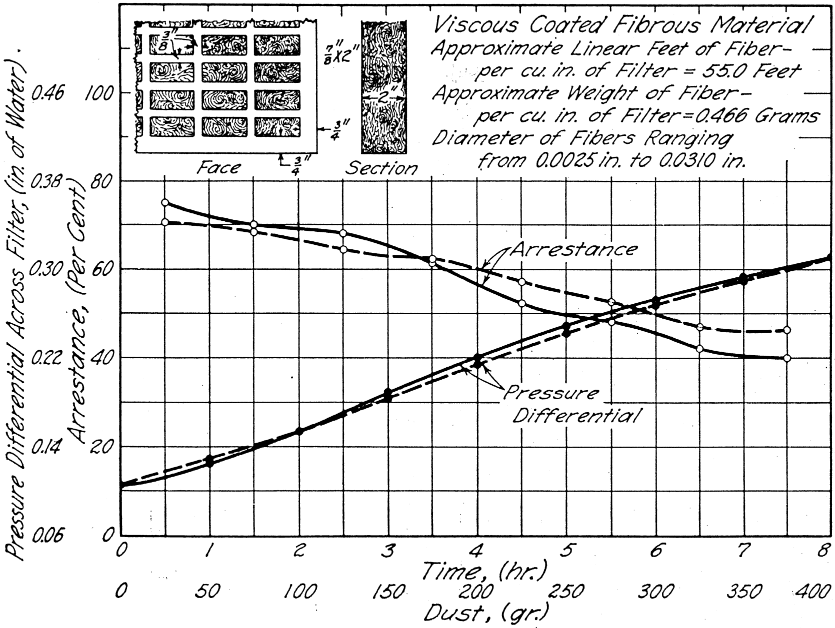

Fig. 5. Results With Viscous Coated Fibrous Filter

the fibers were more loosely packed than in filter No. 5; the arrestance and pressure drop were also less for filter No. 6. In general it may be said that for the same air velocities through the filters the arrestance depends upon the fiber size and density of pack. The denser the filter the higher the resistance, and also the higher the pressure drop, with a corresponding reduction in the dust-holding capacity.

The velocity of air passing through the filters in Tests 4, 5 and 6 was 250 feet per minute and, as will be shown later, a reduction in velocity would likely have increased the efficiency. Low velocity is one reason for the better showing of the dry filter of Fig. 1, although in this filter air resistance was very high even with the low air velocity. The efficiency of the viscous-coated filter of Fig. 4 compares very favorably with that of the dry filter of Fig. 2, although the air resistance for the filter No. 4 increased much more rapidly than that for filter No. 2. The actual face area of filter No. 2 was, however, nearly twenty times as great as that for filter No. 4.

The curves of Fig. 7 show the results for tests on a cellular type of filter in which the walls of the cells were coated with oil. In this

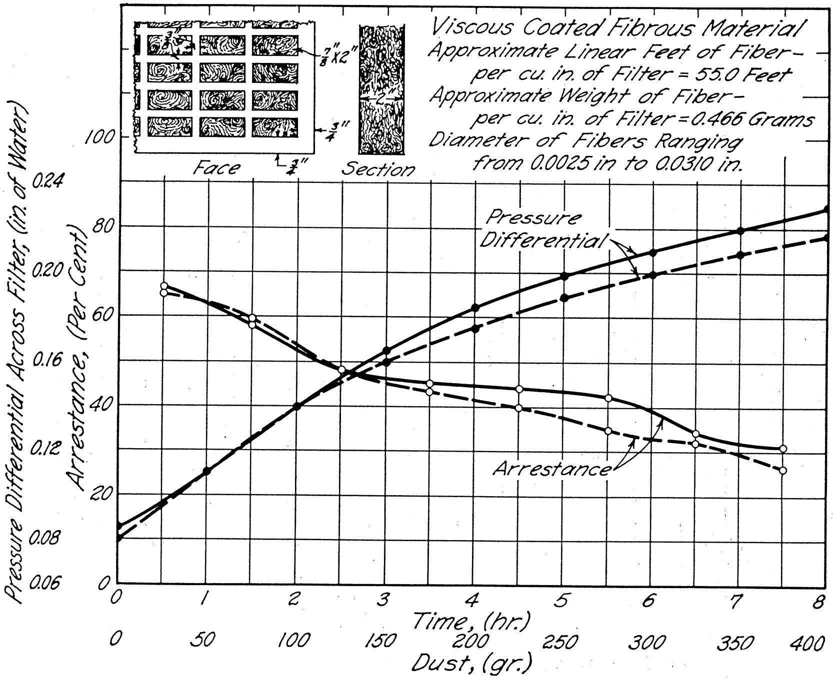

FIG. 6. RESULTS WITH VISCOUS COATED FIBROUS FILTER

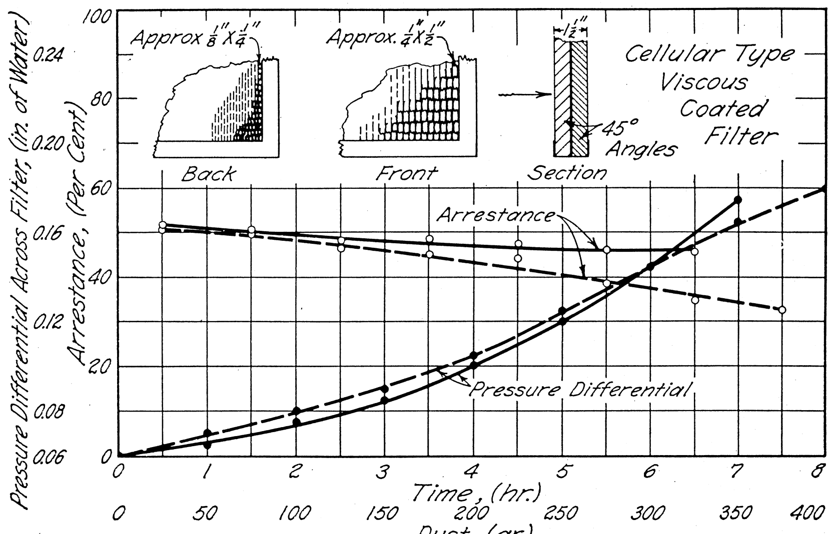

FIG. 7. RESULTS WITH CELLULAR TYPE VISCOUS COATED FILTER

TABLE 1

COMPARATIVE TEST RESULTS OBTAINED FROM AIR FILTERS FOR DIFFERENT MATERIALS AND DIFFERENT AIR VELOCITIES WHEN USING LYCOPODIUM AS A DUST

Test No.	Filter Material	Thickness in.	Air Vel. Thru Filter ft. per min.	Pressure Drop Thru Filter in. water	Arrestance
1	Cotton batting	½	131.4	1.7	85.4
2	Cotton batting	½	73.5	0.73	85.4
3	Cotton batting	½	38.	0.37	92.6
4	½-in. layer of absorbent cotton sprayed with oil	½	34.2	0.40	99.
5	½-in. layer of absorbent cotton sprayed with oil	½	62.	0.62	99.8
6	½-in. layer of absorbent cotton split in two parts and oiled	¼	34.2	0.14	98.4
7	½-in. layer of absorbent cotton split in two parts and oiled	¼	62.	0.40	94.
8	Felt pad, same material as for No. 1		34.2	0.14	89.
9	Felt pad, same material as for No. 1		68.	0.35	59.
10	Same as No. 8 with filter oil added		34.2	0.19	95.
11	Same as No. 8 with filter oil added		62.	0.45	96.
12	⅛-in. felt cloth		66.6	0.42	89.
13	⅛-in. felt cloth		32.7	0.22	96.
14	Same as No. 12, sprayed with oil		34.0	0.22	98.
15	Upholstering moss 4-in. thick, wt = 0.365 lb.		124.	0.07	84.
16	Upholstering moss 8-in. thick, wt. = 0.781 lb., oil added		124.	0.11	98.6
17	Upholstering moss 8-in. thick		69.	0.05	98.8

case the dust particles were taken out by impinging the air against the walls of the cell as the air passed through them. The air resistance of this filter was low, and likewise the efficiency was low. More impinging surfaces would have improved the efficiency and likewise increased the pressure drop.

Some of the variations which may be expected from the different treatments of filter materials are shown in the test results of Table 1. These test results were obtained by using lycopodium powder as a dust in the air. Lycopodium corresponds very closely in size and characteristics with some of the pollens which produce hay fever, and the particular object for these tests was to find the efficiency of the various materials in removing such pollens. The efficiency was determined by taking a dust count of the air entering and leaving the filter by the Smith-Greenberg method. The significant points to be noted in the results are the variations in air resistance and arrestance with air velocities and for different treatment of the fibers. The efficiencies of dry cotton batting filters ½ inch thick for varying air velocities are shown in Tests 1, 2 and 3. From these it will be noted that, as the air velocities vary from 131.4 feet per minute down to 38 feet per minute, the air pressure drop through the filters varies from 1.7 down to 0.37 inches of water, and the arrestance increases from

85.4 to 92.6. This same relation is shown by comparison of Test 4 with Test 5, Test 6 with Test 7, Test 8 with Test 9, Test 12 with Test 13, and in practically every case a reduction in velocity increases the efficiency of the filter and lowers the air resistance. It is thus evident that velocity is a very important factor in the efficiency and operation of filters of this type.

Next consider the effect of spraying the fibers with some type of viscous oil, the other conditions remaining the same. The effect of oil on the cotton filter is shown by comparing Test 4 with Test 3 and Test 5 with Test 2. While the velocities for these comparative tests are not exactly the same, there is evidently but little change in the air resistance due to the oil, although there is a marked increase in the efficiency. By comparing Test 10 with Test 8 and Test 11 with Test 9, in which the same material was used, excepting that it was sprayed with oil for Tests 10 and 11, it is found that the oil gives a small increase in pressure drop and a material increase in efficiency. In Tests 13 and 14 the addition of oil made less of an improvement in efficiency, possibly due to the fact that the initial efficiency was reasonably high, but there was no change in the air resistance for the filter.

Tests Nos. 16 and 17 show the possibilities of obtaining a very efficient and low-resistant filter by the use of some such material as upholstering moss covered with oil or a viscous material. This, of course, is on the same principle as many other viscous-coated fibrous filters, although the moss used in this case was of a fairly fine fiber and not densely packed.

The density of pack determines the air resistance through the filter, controls the efficiency for a given fiber, and affects the dust holding capacity. To get the best combination of these factors many filters are built by progressively packing the fiber to increase the density as the air passes through.

It is possible to get effective cleaning of the air at reasonable cost for power and initial outlay for equipment, but to do this there are certain fundamental requirements which must be observed. First, a filter medium must be selected which is adapted to the particular problem. Second, it must be arranged in the conditioning system in such a manner that it may perform to the best advantage. Third, it must be given some attention in order to keep it in operating condition.

The selection of the best filter for a job is not an easy task. While there are certain types of filters which from a general inspection may be judged to have low efficiency, there will still be those which appear to have all of the requirements of a good filter, and yet show a wide spread in their performance characteristics. Usually the layman will do best to rely upon the judgment and recommendations of a reliable manufacturer of filters or of equipment using filters for the initial selection. The quality of the filter will show up in a short period of operation.

As to the arrangement of the filter, it is necessary first to select an advantageous point in the air stream for taking out the dust. This is usually at the entrance to the conditioning unit, but other arrangements are possible. Previous discussion has shown that low air resistance and high efficiency usually result from low air velocities passing through the filter. For this reason air velocities through the filter are much lower than those for economy in other parts of the apparatus and it is necessary to increase the area of the filter in proportion to other sections of the apparatus. In the dry filter built up of a felt or other thin material this increased area is provided by building it up in accordion plaits as shown in Figs. 1 and 2. By such an arrangement the filter area can be increased to as much as 20 or more times the cross sectional area of the filter, thereby giving low velocities, low air resistance, higher efficiencies, and longer life to the filters.

For those filters which are built in the form of a pack of fibrous materials and do not lend themselves to the construction just described there are at least two ways of obtaining similar results. First, the filter packs, which for average filters are 20 inches square, may be arranged in a staggered position across the duct. By this arrangement the area or air passage through the filter may be effectively increased. Another common method is to expand the area of the pipe in the filter section, thus giving lower velocities.

The necessity for low air velocity and low air resistance depends somewhat on the type of installation. For instance filters are sometimes used in gravity warm-air furnaces. Since the total pressure head causing air circulation is then very low the filter resistance becomes an important item. In some gravity installations the filters are placed at the warm-air register outlets, and, while the filter resistance may interfere with the air circulation through the particular register, it will not seriously affect the other parts of the system. In other gravity installations the filters are placed in the cold air return, and if in this position the resistance is allowed to build up there will be an unbalancing of the air flow through the riser to the different floor levels, and ultimately there may be a reversal of air flow in some of the lower floor ducts.

Lack of attention and upkeep is probably one of the most serious obstacles to successful air filter operation. This is particularly true for the small installations where but little attention is given to the plant. The system takes on dirt which not only lowers its efficiency but also increases the air resistance, and in time may effectively block off all air circulation. The filters are usually installed in such a way that they they are not exposed for inspection without opening at least an access-door to the system, and it is not an uncommon thing to find the filters of a perfectly good installation so filled with dirt and dust as not only to reduce the efficiency of the filter but to throw the whole air conditioning plant out of commission. An inspection of many plants will show that rather than replace or clean the filters some of them have actually been removed in order to reduce the air pressure drop.

In conclusion, there seems to be no argument against the necessity of using air filters in many installations. The dirt and dust in the air is apparent, their effect on both property and health have been established, and the possibility of cleaning them out of the air has been demonstrated. There are many good filters available which, if properly installed and maintained, will do a first class cleaning job. It is, however, necessary to select a filter suitable to the job, and to give reasonable attention to cleaning or replacing the filter to keep it in an efficient operating condition.

VI. AIR CONDITIONING EQUIPMENT

E. L. BRODERICK*

In the last few years the term "Air Conditioning" has been used with such a variety of meanings that any attempt to group or classify the equipment now available in this field may logically begin with a definition of the term.

Committees and individuals who have formulated definitions for complete air conditioning have included or omitted a variety of minor details, but all of them without exception have agreed on the following important factors:

For winter only................heating, humidifying
For summer only..............cooling, dehumidifying
For both winter and
summer....................cleaning, circulating, reduction of odors

Any plant which controls all seven of these factors properly deserves to be called an air conditioning system.

1. *Separate Unit Systems.*—In this latitude, where all homes must be heated, it is convenient to think of any air conditioning system as being built in connection with the heating unit. Following this idea the various types of heating systems may be examined to see what equipment would be required to provide for the other factors.

Since the types of heating systems are fully described in Paper No. VII at this Conference only the identifying features need to be mentioned here. A steam or hot water heating plant which places one or more radiators in each room will be first considered.

Systems of this general type cannot provide for cooling in summer unless the radiators are replaced by units similar to the one shown in Fig. 1. In this unit the two fans circulate the air over the finned coils which occupy the upper section. The small one shown at the top is the heating element, and the larger one shown below it is the cooling coil. Separate sets of pipes are provided to carry steam or hot water in winter and cold water or a refrigerant in summer from central basement plants to the respective elements. With forced air circulation in winter one unit may replace more than one radiator.

The problem of humidifying the air is solved by using water sprays located in the lower part of the unit, but dehumidification can be accomplished only by the action of the cooling coils in condensing the moisture out of the air. A drain pan at the bottom of the unit

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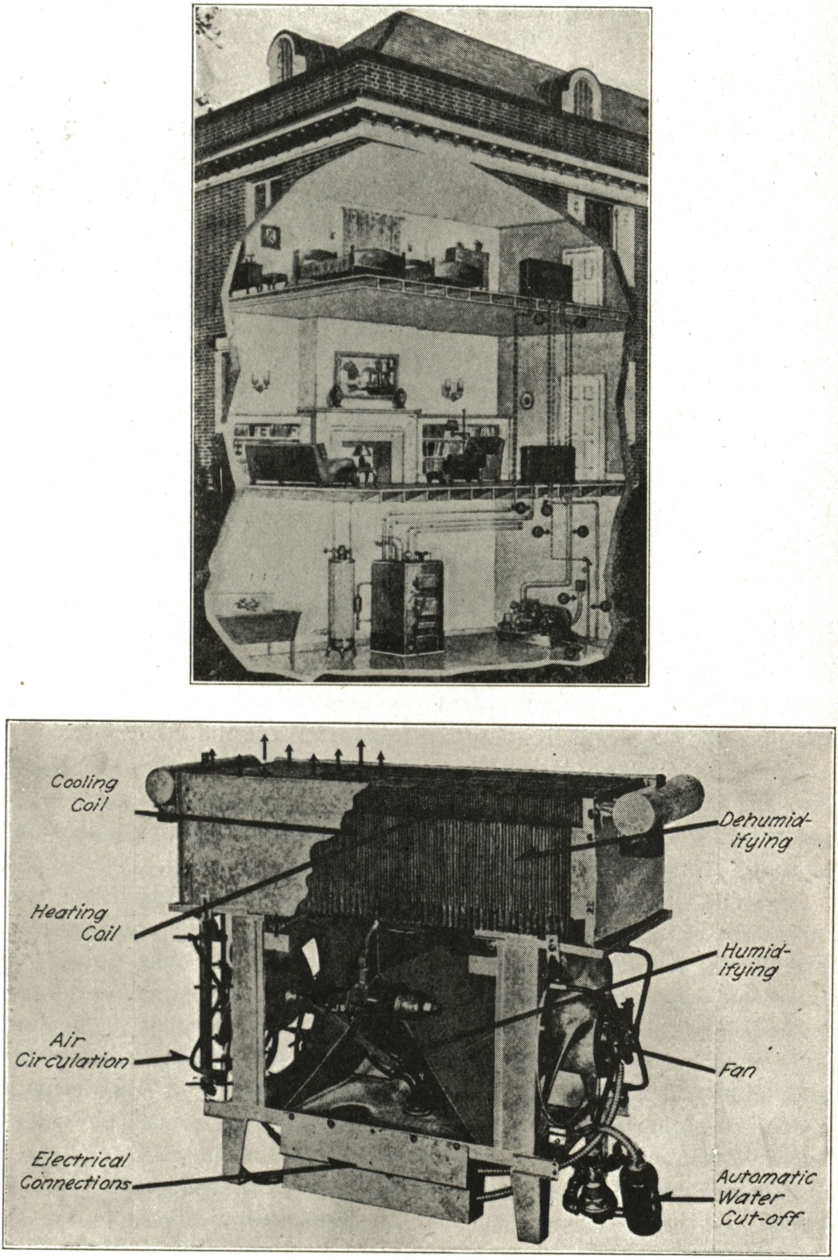

FIG. 1. ROOM UNIT AIR CONDITIONING SYSTEM

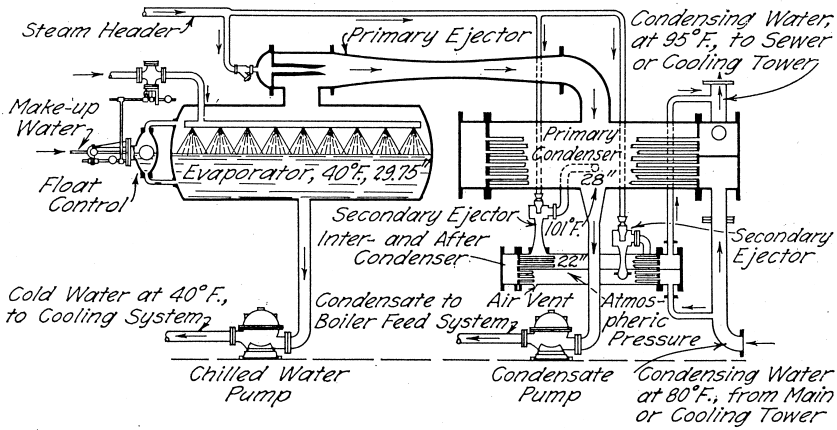

FIG. 2. FLOW DIAGRAM, STEAM-JET REFRIGERATION

collects the excess spray water in winter and the condensed moisture in summer.

Filters may be placed in each unit for cleaning the air, and outdoor air, which is the usual means used for the reduction of odors, may be introduced through ducts into the back of some or all of the individual units.

When provision has been made, as outlined, to control all of the seven principal factors, a steam or hot water heating plant becomes a complete air conditioning system.

The mechanical refrigeration system of cooling a home, indicated in Fig. 1, which uses direct expansion of a refrigerant in the individual units, is a well known and very successful system but it is not the only cooling method available. Cold water circulated through the room units can be used as a cooling medium and it may be drawn from some natural source or may be artificially cooled. A mechanical refrigeration system may still be retained to cool the water which is circulated through the room units, but the particular cooling system selected for description here is the steam ejector refrigeration system since it works hand in hand with steam heating. Other cooling methods will be explained later.

A steam ejector refrigeration system makes use of the fact that water will boil at a temperature of 50 deg. F. or lower when subjected to an absolute pressure which is so small that it approaches a perfect vacuum. Under these conditions a small amount of water evaporated at such a low pressure will cool the remaining water to the boiling temperature. This process, as illustrated in Fig. 2, provides

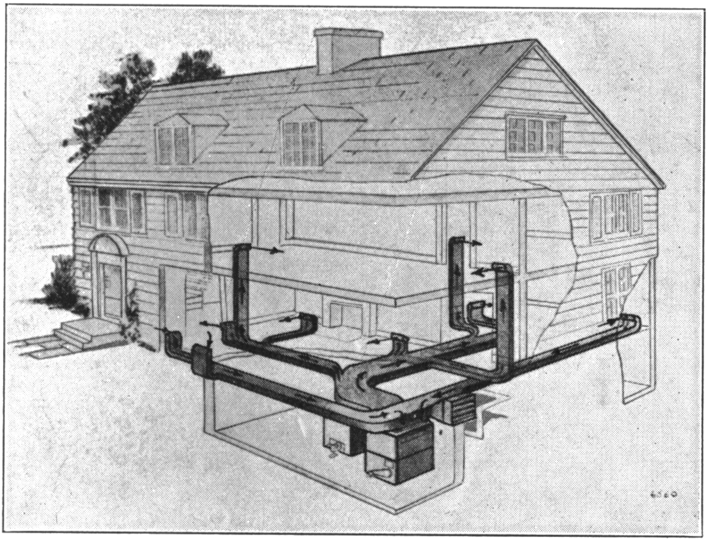

FIG. 3. CENTRAL AIR CONDITIONING SYSTEM FOR RESIDENCE

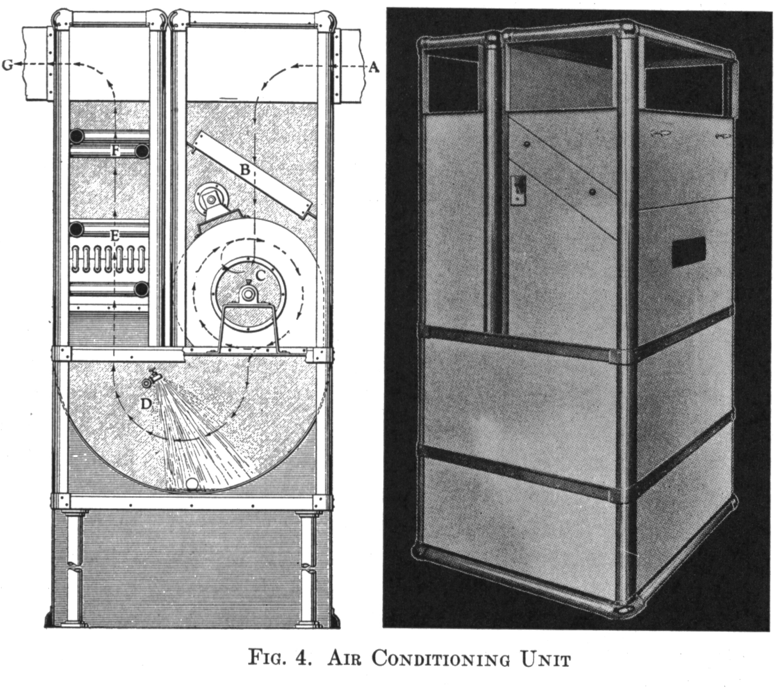

FIG. 4. AIR CONDITIONING UNIT

for the boiling of the water in the evaporator at a temperature of 40 deg. F. and a vacuum of 29.75 inches of mercury. The cold water, which has been chilled by boiling away part of the water in the evaporator, is pumped to the room units by the chilled water pump and returns through the sprays to the evaporator, where it is again cooled by evaporating another small amount. The water that is thus evaporated may be replaced from any source. Condensate is frequently used, since it is practically free from impurities.

The vacuum required for continued operation of the evaporator is maintained by the primary ejector shown in the figure. The jet of live steam travels through its expanding tube drawing air and vapor with it, which produces the vacuum. The mixture of steam, air, and vapor from the ejector is discharged into the primary condenser where the condensing water liquifies most of the steam and vapor for removal through the condensate pump. A small amount of steam and vapor still remains with the entrained air as it is carried through the inter and after condensers and discharged to the atmosphere.

Formerly, cooling systems of this type were used only in large buildings. Now, however, there are units available with a capacity as low as two tons, which is small enough for a great many of our present day homes. The steam pressure required for operating the ejector has also been reduced to a value reasonable for use in a home.

2. *Central Plant Systems.*—The alternative to the system which uses a separate unit in each room is a duct system which circulates the air through a central conditioning plant. The cutaway diagram, Fig. 3, shows the path of air from the house through the return ducts to the basement with a special connection for introducing air from outdoors. The mixture is drawn through the air conditioning unit and sent back to the individual rooms through the distributing ducts. Forced air circulation must always be provided in order that cool air may be supplied to the rooms in summer.

A system of ducts for collecting and distributing the air which is to be conditioned can be used with any kind of a central air conditioning plant located in some out of the way place such as a basement. The unit shown in Fig. 4, is typical of the complete units which are available. Six of the factors necessary for complete air conditioning are provided for. To satisfy the seventh factor requiring the reduction of odors a duct must be added through which outdoor air may be introduced. The filters for cleaning the air are located as shown at *B* in the diagram, and the fan, *C*, handles the air circulation. In winter, the water spray, *D*, humidifies the air and the coil, *F*, does

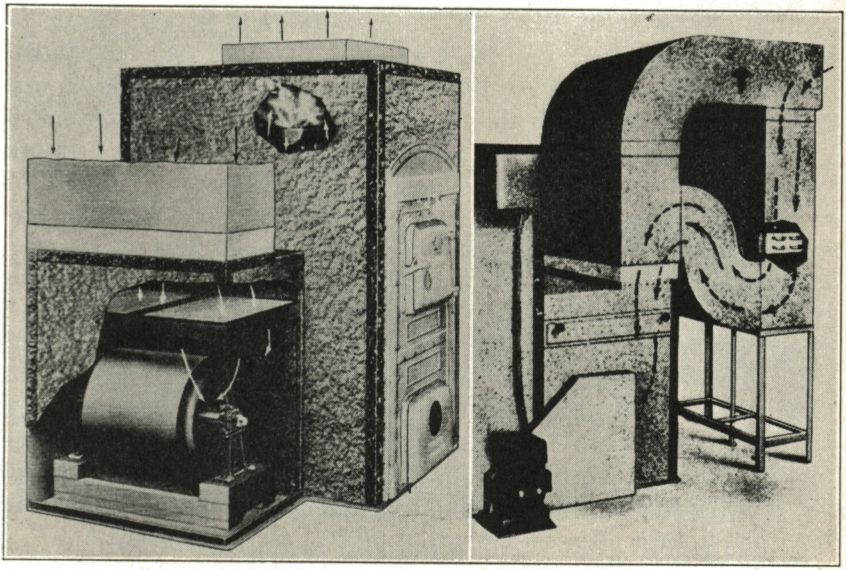

FIG. 5. FURNACE AIR CONDITIONING UNIT

the heating. For summer conditioning the cooling coil, *E*, provides the cooling, and also dehumidifies the air by condensing the moisture on the coil surface.

In most cases this type of unit is intended for use with steam or hot water boilers, but it can be used with a warm-air heating system by omitting the heating coil and attaching the unit to a warm-air furnace. When such a unit is used with a gravity warm-air furnace all of the pipe dampers in the duct system must be set to properly distribute the flow of air. Some changes in the ducts may also be necessary to properly balance the system. If a home has a forced warm-air heating system the furnace fan may be removed and a unit of this type installed, but it is more likely that the forced warm-air heating plant will be left intact, and that equipment for summer air conditioning will be added.

A typical arrangement of a complete air conditioning system based on a forced warm-air heating system is shown in Fig. 5. Six of the seven air conditioning factors are again controlled in the unit and the seventh one, reduction of odors, must be taken care of, as before, by adding fresh air at some point in the return ducts. In the unit shown, the filters and fan provide for the cleaning and circulation, the furnace does the heating for winter air conditioning, and the

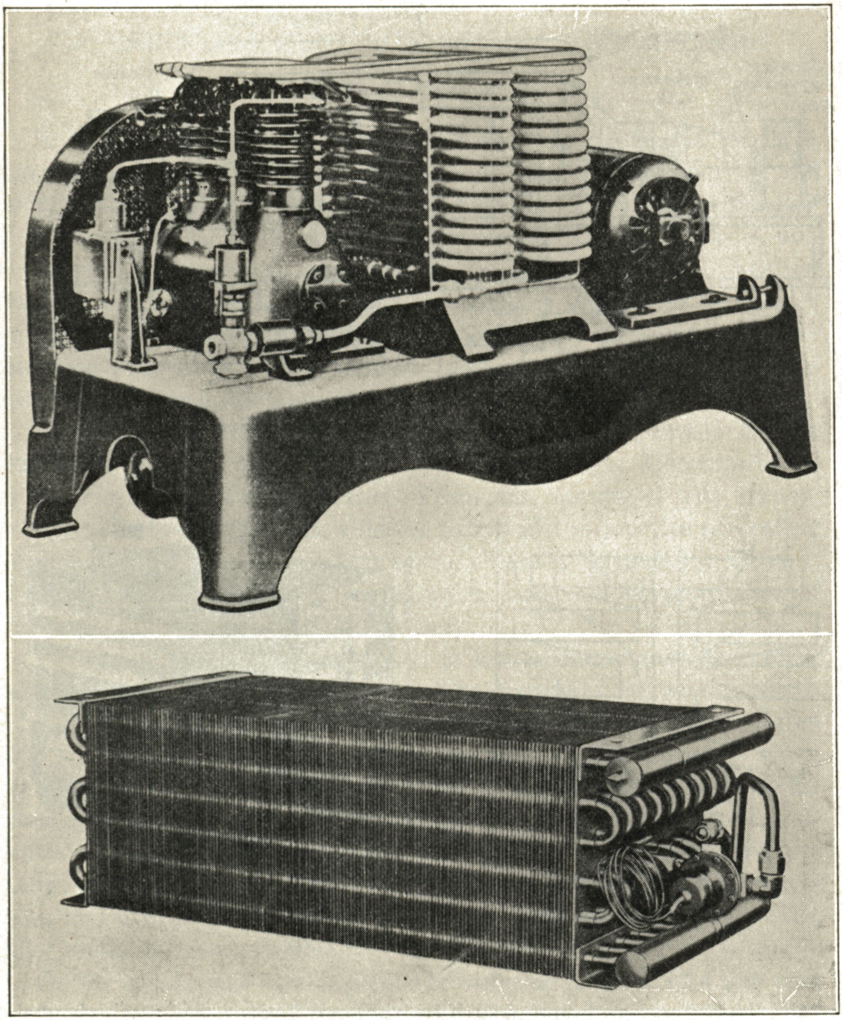

Fig. 6. Mechanical Refrigeration Equipment for Air Conditioning

humidifier near the top of the furnace casing furnishes the water vapor which must be added to the air. The cooling coil again takes care of both of the summer factors, cooling and dehumidification. The by-pass feature built into this unit allows the air to be circulated around the cooling coil when it is not in use.

Probably the most common method of providing cooling for a home employs a mechanical refrigeration system using equipment as shown in Fig. 6. The electric-motor-driven compressor is installed

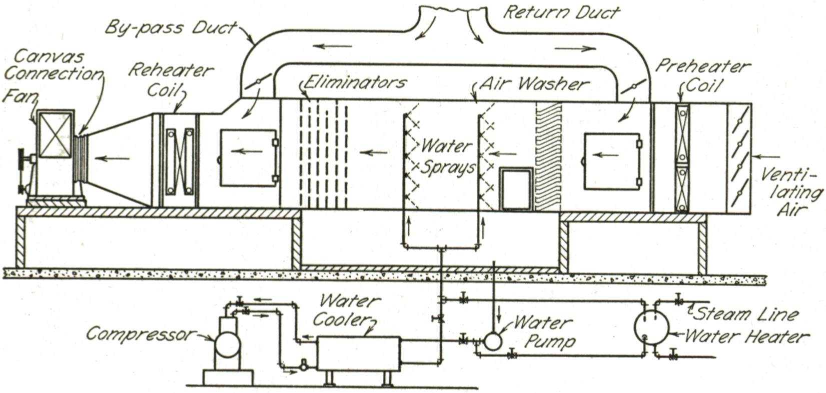

FIG. 7. ELEVATION OF TYPICAL CENTRAL PLANT AIR CONDITIONING UNIT

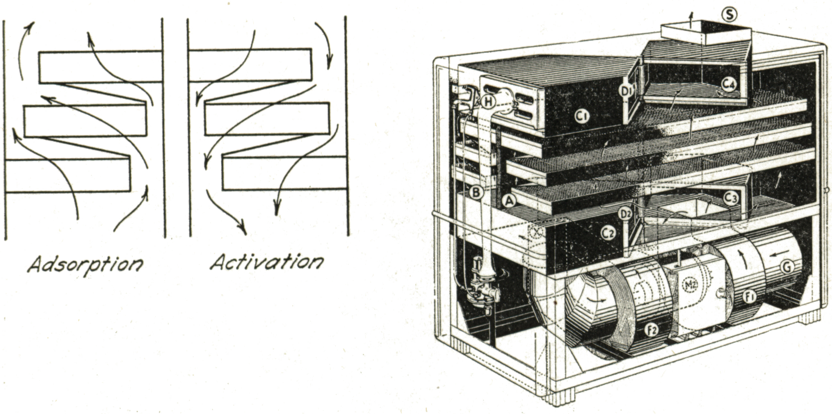

FIG. 8. DEHUMIDIFYING UNIT

in the basement and connected to the evaporator coils which are located at some point in the air conditioning system. The compressor is usually water cooled, as shown in the figure, but there are machines available which are partially or entirely air cooled. Among the many refrigerants used in these systems are dichlorodifluoromethane, methyl chloride and ammonia.

In explaining the operation of these refrigeration systems it is convenient to begin at the compressor. The compressor itself is really a pump which draws the refrigerant from the evaporator in the form of a gas and compresses it. The gas is pumped directly into the condenser where the action of the cooling water removes enough heat to liquefy it. From the condenser the liquid refrigerant moves under pressure to the evaporator passing through an expansion valve just as it enters the coil. In the evaporator, as a result of the reduction in pressure, the refrigerant boils. The heat required to boil the refrigerant is taken from the air passing over the coils, and produces the cooling effect. The evaporated refrigerant returns to the compressor on the suction stroke and thus completes the cycle.

A third scheme for air conditioning with a central plant unit makes use of an air washer, Fig. 7. In winter some air is brought in from outdoors, passed through the preheating coil and mixed with the recirculated air. The mixture is drawn through the warm water spray in which it is humidified and partly cleaned. The cleaning is completed on the eliminator plates which also remove any entrained moisture. The air is then heated to the proper temperature by the reheating coil and delivered to the house by the fan. Summer air conditioning depends entirely on a cold water spray. The mixture of fresh air and recirculated air passing through the water spray is cooled and will be dehumidified if the temperature of the water leaving the washer is below the dew point of the entering air. In this case also the eliminator plates remove the entrained moisture and effect most of the cleaning.

In all of the equipment mentioned up to this point, the problem of dehumidifying has been definitely related to the cooling, and no method has been shown for independently controlling this factor. There are units available which will remove moisture from the air by adsorption using solids such as silica gel and activated alumina, or liquids such as a solution of lithium chloride. The dehumidifying unit shown in Fig. 8 used a solid as the adsorbing substance. The unit is divided longitudinally into two separate sections which are in service alternately. While one section is adsorbing moisture from the air to be conditioned, the hot gases from the gas burner, *H,* are passing through the other section re-activating the adsorbing substance. Dampers, automatically controlled, reverse the activation and adsorption at regular intervals and thus the dehumidifying process becomes continuous. In an air conditioning system, it is general practice to pass only part of the air through the dehumidifying unit. One convenient method removes moisture only from the fresh air admitted for the reduction of odors.

3. *Unit Coolers.*—In recent years the high cost of complete air conditioning systems has fostered the development of equipment which is entirely separate from the heating system and provides only

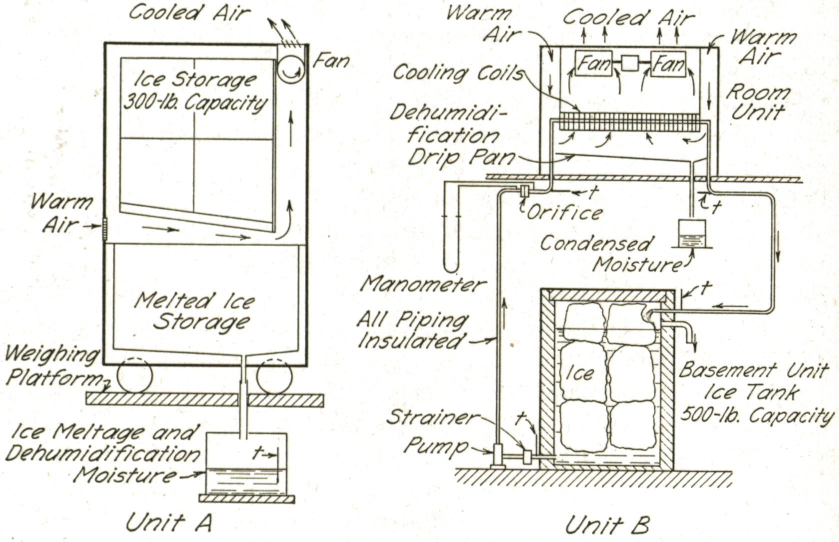

FIG. 9. DIAGRAM OF COOLING UNITS AND TEST ARRANGEMENT

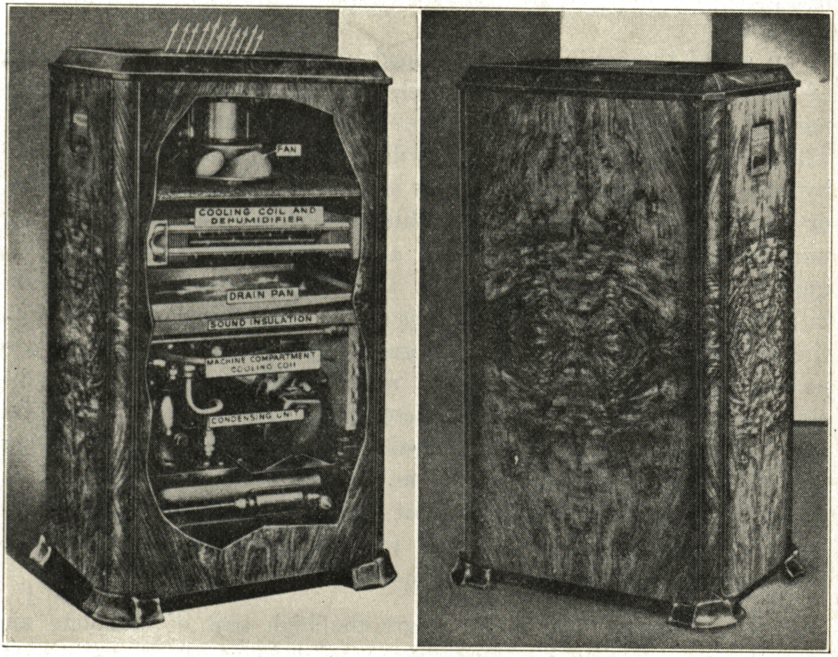

FIG. 10. SELF-CONTAINED ROOM COOLER

for some form of cooling in summer. These units have been used for various purposes, such as partial cooling of a large space or complete cooling of a single room, and have employed nearly all of the cooling mediums common to the complete systems.

Figure 9 presents diagrams of two commercial units which were used in the Warm-Air Heating Research Residence during 1932 and 1933. Unit A cools and dehumidifies the air by direct circulation over ice. It has the advantage of being movable and has a fairly low first cost, but there is no provision for adding fresh air to the cooled space and the problem of refilling the unit with ice and removing the meltage may become troublesome. Unit B employs chilled water for cooling. This water may be chilled with ice or obtained from some other source. The temperature of the water available will determine the capacity of the unit, and also the amount of dehumidification for each of three selective fan speeds. The unit is necessarily stationary, but it does not have the recharging problem presented by unit A. A fresh air inlet could be built into the back of the cabinet.

One of the most common types of room unit coolers, Fig. 10, is designated as self-contained because all of the machinery for conditioning the air is built into one cabinet. Since a mechanical refrigeration system is used to cool the air the description given for the equipment shown in Fig. 6 applies here. The compressor, condenser, and evaporator shown in Fig. 6 are labeled condensing unit, machine compartment, cooling coil and cooling coil respectively, in this figure. Air from the space to be cooled enters at the back of the cabinet, passes over the evaporator coil where it is cooled and dehumidified, and after passing the fan is delivered to the room through the top of the cabinet. With the air inlet at the back of the unit it is possible to make a connection to the outdoors and to admit any desired percentage of fresh air. Some units are built to be placed in front of a window and are arranged to include a fresh air intake connection which uses part of the lower sash space. If the fresh air intake is equipped with a filter as a final refinement, the room cooler may properly be called a summer air conditioning unit.

Units of the self-contained type are, as a rule, not portable. The pipe connections, namely, cooling water inlet, cooling water outlet, and a drain for moisture condensed from the air, make a permanent location necessary. With three pipes already in place the electric wiring will most likely be placed in conduit.

Efforts to make these units more portable have progressed in different directions. The unit shown in Fig. 11 uses a flexible hose specially

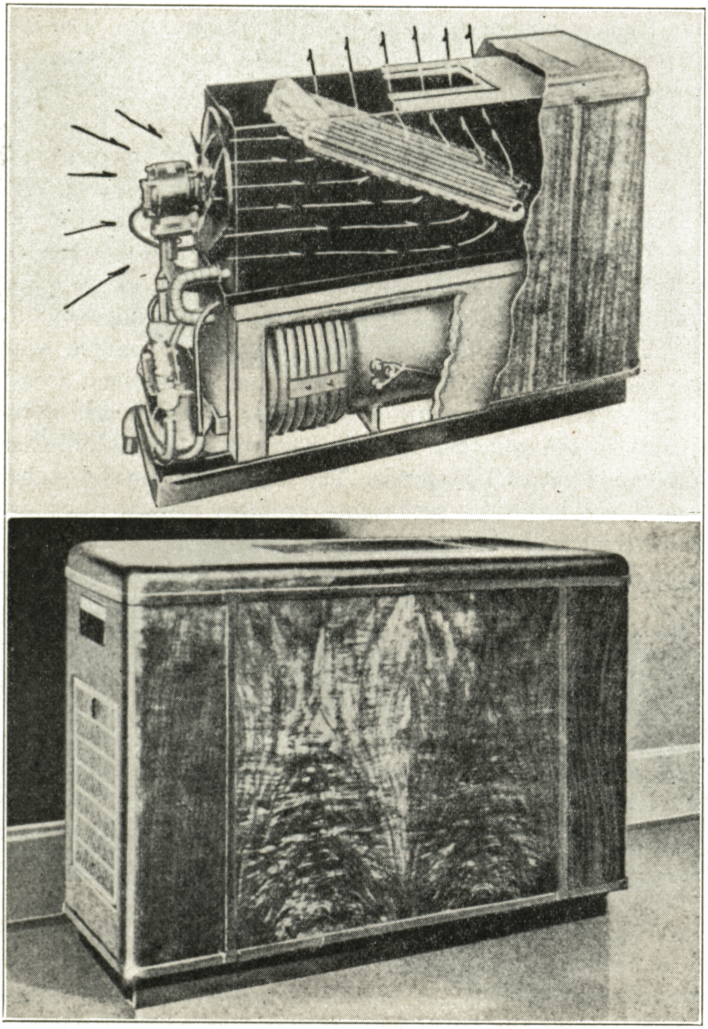

Fig. 11. Unit Room Cooler

molded with separate tubes for inlet water, outlet water, and moisture condensed from the air. With no filters and no provision for fresh air this is strictly a room cooling unit with the dehumidifying and circulating features automatically included.

A second attempt to secure portability has resulted in the development of an air cooled unit, Fig 12. Air cooling for a compressor

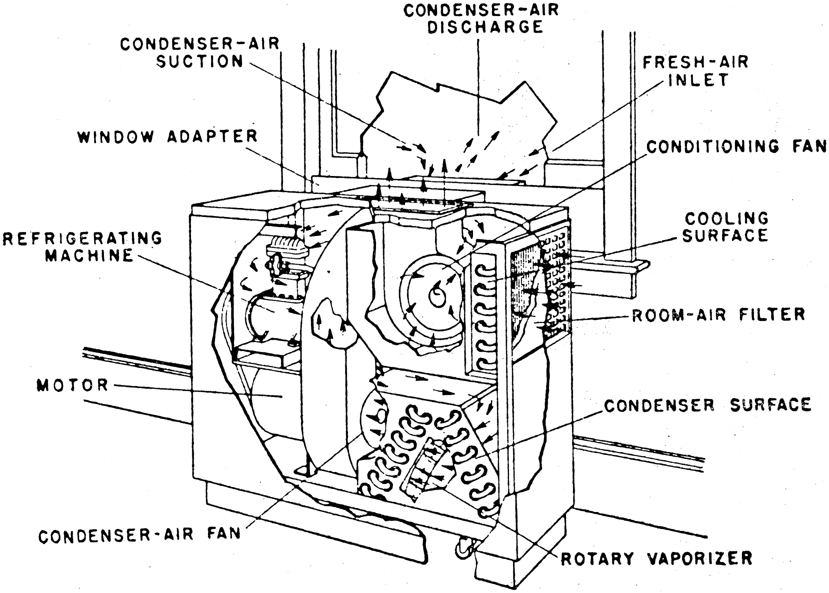

FIG. 12. PORTABLE SELF-CONTAINED AIR COOLED CONDITIONING UNIT

and condenser requires forced circulation which is provided by the condenser-air fan.

Another feature incorporated in this unit is a means of removing the moisture condensed from the air without using a drain pipe. The moisture condensed on the surface of the cooling coil drains into the rotary vaporizer where it is evaporated and carried outdoors by the air circulated over the condenser.

The air cooling compartment is similar to the corresponding sections of other units. Air from the room is circulated over the cooling coil which cools and dehumidifies the air. Some fresh air is admitted from outdoors and a filter is placed at the room air inlet. This accounts for all five of the factors required for summer air conditioning and provides a unit which is really portable.

Cooling units in homes are usually located on the floor, but in small shops where floor space is more valuable they are frequently suspended from the ceiling. The unit shown in Fig. 13 uses the double fan to circulate air over the coils which provide for cooling and dehumidification. Filters are located at the air inlet at the back of the casing which makes them readily accessible. The fresh air necessary

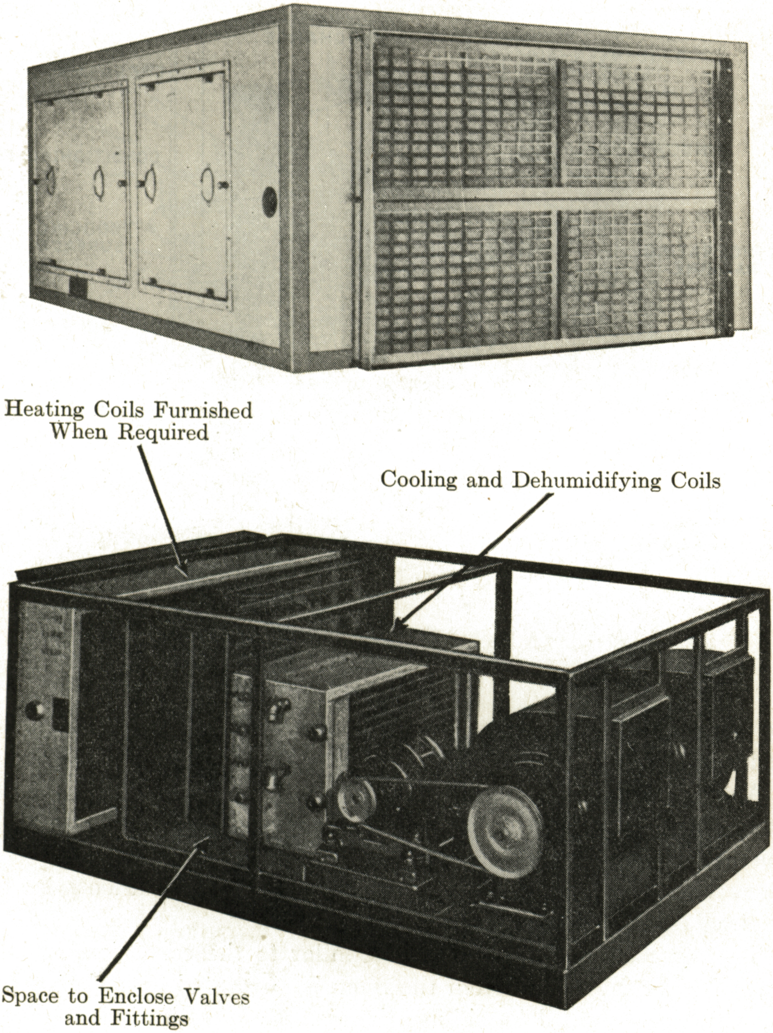

FIG. 13. SUSPENDED AIR CONDITIONING UNIT

to complete the work of conditioning the air in summer must be supplied through ducts connected to the inlet. The heating coil shown is capable of doing part of the winter air conditioning, so if a humidifier, which is available, is installed, the unit becomes a complete air conditioning system.

4. *Night Air Cooling.*—One means of securing partial cooling in a home depends on a schedule of operation for the windows. Keeping

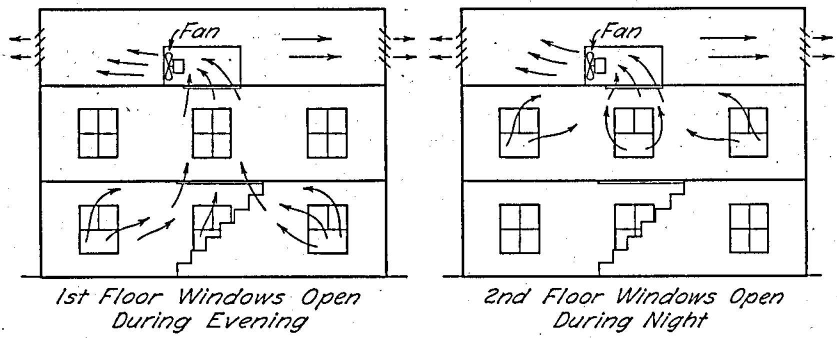

FIG. 14. NIGHT AIR COOLING SYSTEM

the windows open at night will allow the circulation of night air which will pre-cool the structure and keeping the windows closed during the day will help to retain the effect of the night cooling.

If a system of this kind is used, some opening should be provided through the attic of the house which will be operated with the windows to give a chimney action up through the structure during the night cooling. This arrangement may provide cooling which is very satisfactory in mild weather, but during long hot spells the effectiveness is lost. Also, it is very difficult to keep the house free from objectionable odors when it is closed during the day and no fresh air is provided.

Night cooling systems may be made more effective by installing an attic fan, Fig. 14, which will increase the air circulation, and give the structure a cooler start into a hot day. The schedule of operation, called for in the figure, which circulates the air through the first-floor rooms during the early evening and through the second-floor rooms during the rest of the night provides maximum comfort for the occupants of the house. A fan in a furnace, or basement air conditioning unit, may be used in the same way if some arrangement is made to admit outdoor air to the fan at night, but the quantity of air circulated is usually much smaller and the cooling effect is reduced.

If a night air cooling system is used alone it is an economical means of providing partial cooling, but if it is used as a means of reducing the operating time of a cooling plant it is possible that the saving made on the main plant will be offset by the cost of operating the fan at night. Natural ventilation of an attic during the day is very worth while in helping to keep a house cool, but the value of forced circulation is doubtful.

Conclusion.—No attempt has been made in this discussion to select any one air conditioning system as the best. The seven principal factors required for complete air conditioning were stated in the beginning, and the rest of the paper has been devoted to the task of showing how they are controlled in different types of systems.

VII. ESSENTIAL FEATURES OF HEATING SYSTEMS

By P. E. Mohn*

In Illinois, during seven months of each year, heating is the most important phase of air conditioning. Accepting the necessity for the use of heating systems in the houses in the state, it will be the purpose of this paper to show the existing distribution of heating systems, and to present the essential elements of the several types of heating systems available. While each of the systems considered is equally capable of performing its heating function, it is proposed to examine the more commonly used systems to determine their adaptability to year-round air conditioning in the accepted sense of the term.

During 1934 a Real Property Inventory was conducted by the Bureau of Foreign and Domestic Commerce of the Department of Commerce of the United States. From the results of this survey, data have been compiled showing the extent of the practice of heating residential buildings. A summary of these data is presented in Table I. It should be noted that the inventory is based on urban (towns of 2500 or over) residential dwellings only.

For the purposes of tabulation, it is convenient to classify heating systems according to the medium used to distribute the heat throughout the house. The distribution of such systems throughout the United States is unimportant because a large part of the country where central heating is unnecessary is included in the totals. It is of interest to examine the record of Illinois. In the total number of residential dwellings heated by warm air, Illinois rates first among the states, eleventh in steam and vapor systems, fifth in hot water systems, and fifth in the number of residential dwellings heated by stoves, other methods, and unheated.

Of the residential dwellings in the state of Illinois, 65.4 per cent are heated by warm air, 2.2 per cent by steam and vapor, 4.7 per cent by hot water, with 27.7 per cent unclassified or unheated. As might be expected, over 90 per cent of residential warm-air and hot-water heating systems are found in one- and two-family dwellings, while 75 per cent of the steam and vapor systems are in this classification.

It is interesting to note the relative distribution of the three types of heating systems in Illinois as compared with that in the United States as a whole. Of residential dwellings heated by these three types of systems 90.4 per cent are heated by warm air in Illinois, and 73.2 per cent in the United States. Similar comparisons for steam and

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TABLE 1
DISTRIBUTION OF HEATING SYSTEMS IN URBAN RESIDENTIAL DWELLINGS*

Types of Heating System	Warm Air	Steam and Vapor	Hot Water	Stoves, Other Methods, and No Heat
Total in the United States (exclusive of New York City)	6 170 801	1 249 917	1 001 923	7 246 982
Percentage of total in the United States	39.4	8.0	6.4	46.2
Illinois' rank among the States in number of plants	First	Eleventh	Fifth	Fifth
Percentage of total plants in the United States located in Illinois	15.1	2.5	6.7	5.5
Number of plants in Illinois	933 721	31 410	67 102	395 475
Percentage distribution of plants in Illinois	65.4	2.2	4.7	27.7
Percentage in single-family houses	77.3	65.8	79.5	66.6
Percentage in two-family houses	18.5	8.8	10.7	18.0
Percentage in apartments	0.4	12.7	1.4	0.8
Percentage in other residential buildings	3.8	12.7	8.4	14.6
Percentage distribution of three major types of heating systems in Illinois	90.46	3.04	6.5	....
Percentage distribution of three major types of heating systems in the United States	73.2	14.9	11.9	....

*Data compiled from "Heating & Ventilating"—Nov. 1934 and Sept. 1935.

vapor heating are, Illinois 3.04 per cent, United States 14.9 per cent, for hot water heating, Illinois 6.5 per cent, United States 11.9 per cent.

The real property inventory showed that 62 per cent of residential structures are over 15 years old, while 16.3 per cent are in need of repairs. These percentages are approximately correct for the United States as a whole, and for the state of Illinois. Hence, it might be presumed that approximately one-half of existing heating systems are out-moded in the light of more recent developments. All existing heating systems should be examined with the view of changing them to conform to present standards.

In discussing heating systems greater attention will be paid to the warm-air system, due to the predominance of this type of system in the state, and to the fact that a major portion of the heating research work at the University of Illinois has been confined to warm-air heating.

In considering the warm-air heating system, most attention will be paid to the gravity-flow warm-air system because of its widespread use. Figure 1 shows a cast-iron, gravity-flow warm-air furnace with a ring-type radiator. The parts of the furnace are shown and labeled.

For those individuals whose homes are equipped with gravity warm-air heating systems, an excellent check on the system may be had by comparing it with the Standard Code for the installation of gravity warm-air heating systems. This Code is a standard guide for the installer, set up by the National Warm Air Heating and Air Con-

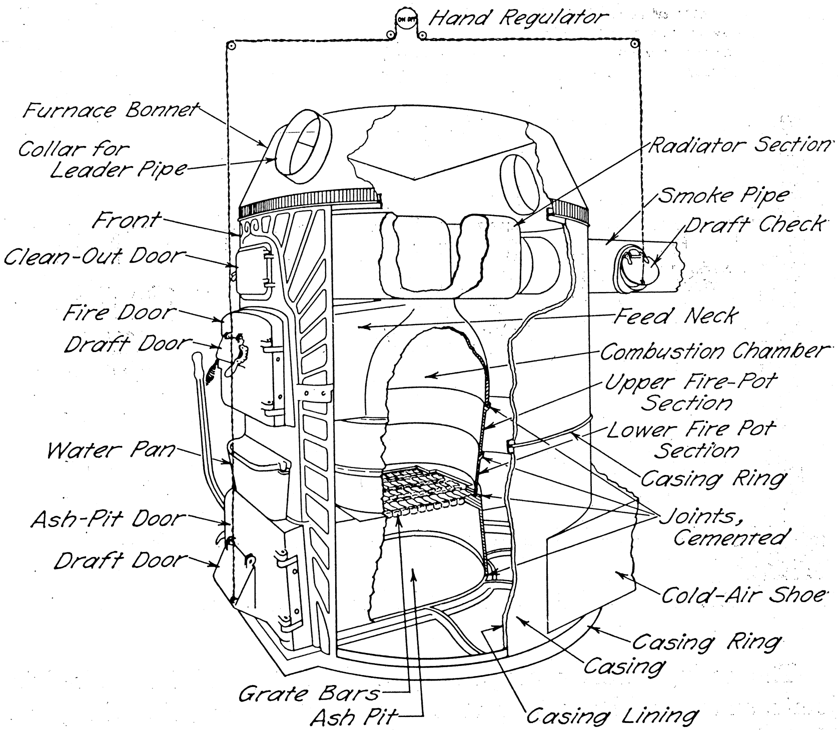

FIG. 1. SECTIONAL ELEVATION OF CAST-IRON GRAVITY-FLOW WARM-AIR FURNACE

ditioning Association, the American Society of Heating and Ventilating Engineers, and the National Association of Sheet Metal Contractors. Systems installed before the adoption of the Code should be checked against Code requirements. More recently installed systems should be checked too, since the rigors of competitive bidding sometimes result in laxity in adherence to the Code.

Some outstanding factors easy to check about your warm air furnace are:

The casing ring should be level, rest on a non-inflammable surface, and be grouted to the floor with neat cement to form a dust- and air-tight joint.

The joints requiring furnace cement should be well filled and absolutely gas tight. Because of frequent changes in temperature the cement does crack and cause leaks in time, and should be renewed periodically.

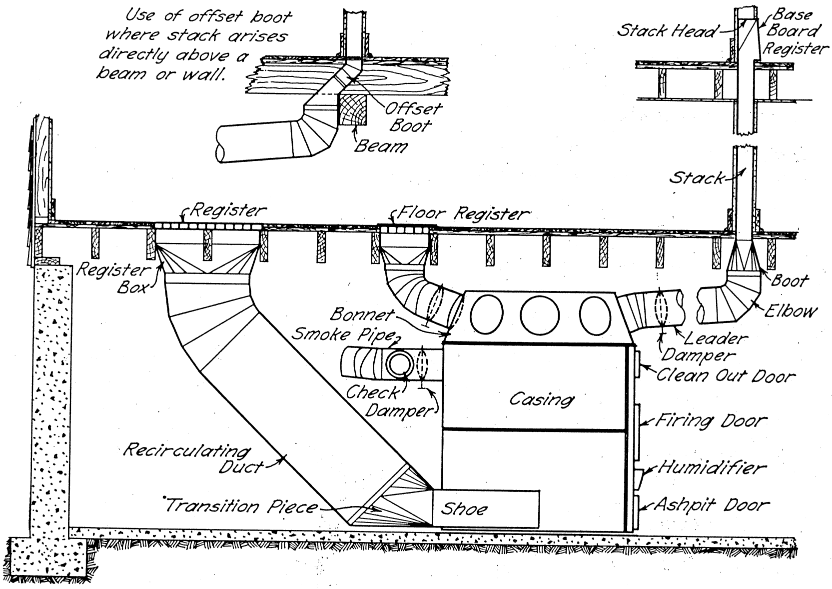

FIG. 2. GRAVITY-FLOW WARM-AIR FURNACE AND CONNECTIONS

The casing, usually of galvanized sheets, should have air-tight joints, and should be lined from its top down to the level of the grate. Tests have demonstrated the effectiveness of this lining in reducing heat losses from the casing.

If the top of the casing is less than 12 in. from a combustible floor or joist, a protecting metal shield should be placed 2 in. below the combustible material. The top of the casing should never be less than 6 in. from combustible material.

The bonnet must be high enough so that leaders may be attached without ovaling.

A water pan or humidifying device should be used. The water pan shown in Fig. 1 illustrates one of the futile locations sometimes used. In this location the water pan has little circulation of air over it and the location is comparatively cool. While most water pans are inadequate for the most severe conditions, a water pan location in the dome is better. Similarly, a pan with a large water evaporating surface is better than one having a small water surface.

The cast-iron furnace shown in Fig. 1 is representative of one type of warm-air furnace. Steel furnaces of welded or riveted construction are also used. Steel furnaces are free from the joint leakage found in cast-iron furnaces, but may be injured due to corrosion or warping caused by overheating. First cost is often the controlling factor in the choice of a furnace between the cast-iron and steel types.

Figure 2 presents an elevation of the basement portion of a gravity warm-air plant, and shows the location of further points to be checked.

The leader pipes should be of bright tin, with air- and dust-tight joints. Tests made at the University of Illinois have demonstrated that the bright tin surface is a better insulator when bare than after being covered by an asbestos paper cover unless at least eight thicknesses of the paper are used.

All warm-air pipes or leaders should have an upward pitch not less than 1 in. per running foot, away from the furnace casing.

If a warm-air pipe passes within 1 in. of wood, the wood should be covered with asbestos paper.

Warm-air pipes should be insulated if they pass through unheated spaces.

All joints, such as those between registers and stacks, should be sealed.

A cross damper, supported at each end, should be placed in each leader pipe.

Recirculating cold-air returns must have a total area, at all points, equal to or greater than the sum of all leader areas.

All boots and shoes should be of stream-line construction.

If spaces between joists are used as cold-air ducts, all wood surfaces should be lined with sheet metal, and the bottom should extend not less than 2 in. below the joists. Connections to such ducts should be made with transition fittings.

The face area of grilles should be at least equal to the duct area.

If vertical cold-air registers are used, any part above 14 in. from the floor is not considered as being effective.

The smoke pipe must be short and have a check damper.

The sizes of grates, leaders, and stacks may be checked by the application of simple arithmetical methods if Code directions are followed.

Tests have shown the most advantageous ratio of free area of casing to leader pipe area to be 1.36 for maximum furnace capacity. This ratio is useful in checking the casing and furnace of the installation.

In the use of the Code in checking a heating system, a local installation will serve as an excellent example. Built some years before the existence of the Code, an examination reveals a bedroom served by a stack placed in an outside wall and without proper insulation. The stack area is 54 sq. in., the leader area 78.54 sq. in., having the limiting Code ratio of 70 per cent. By Code methods, the room requires a leader area of 93 sq. in. plus 15 per cent for unusual exposure, or 107 sq. in.; there is, therefore, a deficiency in leader size of 37 per cent.

The total area of all installed leaders is 568.6 sq. in. There are three cold-air pipes, with areas at the casing of 540, 113, and 78.54 sq. in., respectively, a total of 731 sq. in., which would be ample. However, the minimum areas in each of these ducts are 286, 113 and 35 sq. in., respectively, a total of 434 sq. in., which is far from adequate. Besides, the smaller one is not used because of its extreme location; both of the others have long runs of unlined spaces between joints, abrupt changes of section, and no stream-lined fittings.

The grouting around the casing ring is partly chipped out, the furnace castings are cracked and leaky, the casing joints are not tight, and the water pan is deep with a small evaporating surface, is inadequate in size, and is poorly located. Originally the water pan was located below the grate level, and although its location has been changed, further improvement is possible. In this plant a new furnace installation is advisable.

Considerable attention has been paid to the gravity warm-air system. Most of the precepts which govern this type of system do not apply to the forced-warm-air system. This type of system utilizes a fan to create air motion. Consequently, by utilizing greater motor power, smaller ducts and sharp turns may be used at the expense of added power. The gravity system requires a centrally-located furnace, and leader pipes are limited to not more than 15 ft. in length; these limits are not imposed on the forced system. Since the ducts of forced systems need not be graded, they may be located between joists, providing more head room. Tight joints, however, are essential with either system. The forced system lends itself more readily to automatic control, and to the addition of other air-conditioning features such as filters, coolers, etc.

Consideration of steam and vapor systems is more complicated because of the large number of variations that may be introduced. From the standpoint of domestic use, the discussion will be confined largely to the simplest and least expensive steam system, the one-pipe gravity-circulating steam heating system.

Figure 3a shows a steam system of this type, with the house dis-

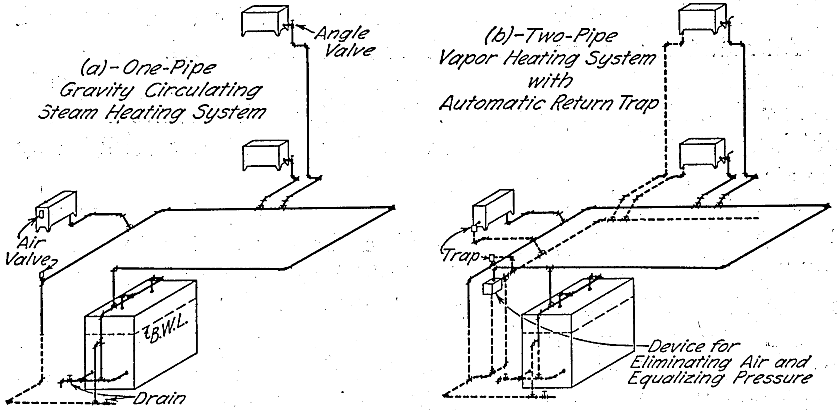

FIG. 3. TYPICAL STEAM HEATING SYSTEMS

solved away. The essence of present day practice is illustrated in the sketch; included are:

A boiler of adequate size

A steam main ample in size, and continuing at constant diameter from the boiler to the point, below the boiler water level, where it is reduced in size as it joins the return pipe

Branches to risers taken out of the top of the main

Branches provided with means for expansion

Each radiator on first and second floors served by its own branch riser

The main graded one inch in 10 feet downward in the direction of flow (away from the boiler)

Each branch riser well graded

Each radiator provided with an angle valve and an air valve; adjustable air valves available

An air valve at the extreme end of each main, at the point where the main drops to the return

Return entering the boiler through an "underwriters' loop"

Drains for boiler and returns

Radiators or convectors (concealed radiators) adequate in size to serve the spaces to be heated

Return pipes grading continuously toward the boiler, at least 1 inch in 20 ft.

In buildings of more than two stories, each riser dripped to the return separately

Steam valves supplying radiators must be fully open or closed.
All radiators must be located above boiler water level.

Systems of this one-pipe type operate at steam pressures slightly greater than atmospheric pressure. Humidifying devices are available which are attached to the radiators at the point usually occupied by the air valve.

A variant of the one-pipe up-feed system shown is the one pipe down-feed system. The main is taken directly to the top of the building, feeding steam into the risers which run downward toward the basement. In this system each riser is dripped to the return.

Some two-pipe gravity systems are still in use. In this system each radiator is drained by a return pipe, each return dropping to the wet return line. In case a high, or dry, return is used, connections to it should be made through a water seal two to three feet deep to prevent reverse flow of steam through the returns.

Figure 3b shows an example of a two-pipe steam system. The vapor system illustrated is characterized by a trap at the return connection to each radiator, a graduated steam inlet valve on each radiator, and a high or dry return. These features are characteristic of most two-pipe steam systems.

Vapor, vacuum, sub-atmospheric and atmospheric systems are variants of two-pipe systems. The steam pressures used are indicated by the name of the system. Each system has its special merits and advocates. The vapor system operates at very low pressures, ounces instead of pounds per square inch. In the vacuum system, a vacuum is maintained in the return line with a positive pressure in the steam line, enabling radiators to be located below boiler water level. The sub-atmospheric system operates with a vacuum in both supply and return, enabling steam temperatures to be varied over a wide range with varying weather conditions. In the atmospheric system the returns are open to atmospheric pressure. This latter system is especially adapted to the use of by-product steam for heating. Orifice systems use orifices at each radiator to proportion the amount of steam each radiator receives. Radiator orifices may be used with any of the steam systems named. Each of these systems has its special uses and advantages. In any steam system, proper pipe and radiator sizing, slope of the pipes in the proper direction, and a continuous downward flow of the returns toward the boiler are essential to proper operation.

Hot-water heating systems may be classified, like warm-air systems, into gravity-flow and forced systems. In the gravity-flow system, flow is caused by the differences in weights of the water in the flow and return risers. The forced system uses a circulating pump in the return line near the boiler to create a positive flow.

Gravity-flow hot-water systems of two-pipe construction are used with either short-circuited or reversed returns. These systems may be either of the up-feed or of the down-feed type. A one-main system may also be used.

An expansion tank is essential to a hot-water heating system. The purpose of the expansion tank is to take care of the variations in water volume with the changes in temperature. An open expansion tank, placed above the highest radiator, or a closed (compressed air) expansion tank, usually located in the basement, may be used with any of the gravity systems. In some rare instances the city water system is used as an expansion tank, with a resultant wasting of heated water. Each of the hot-water systems illustrated is equipped with an open expansion tank. The open expansion tank should have an overflow discharging into a basement drain so that the overflow may be observed. The tank and its vent must be protected from freezing.

Figure 4a illustrates a two-pipe up-feed reversed return hot-water heating system, with an open expansion tank. In this type of system the path of the water from the boiler through a radiator and back to the boiler is of approximately the same length, regardless of the radiator served. Any individual particle of water passes through only one radiator in making the circulation. Each radiator is supplied with hot water at approximately the same temperature. Water is supplied to each radiator at the top and leaves at the bottom of the opposite end. The shut-off valve is placed at the outlet end. Each radiator is equipped with an air valve for venting, to insure that the radiator is full of water at all times. The air valves are of the manually operated type.

Figure 4b shows an overhead distribution system, feeding hot water downward to the radiators. The effect of the radiator connections is to accomplish the same water path as that found in the reversed return system. The radiator connections are the same, except that the air valves are omitted, since all venting naturally takes place from the open expansion tank. If a closed tank is used, the highest points in the system must be vented. The overhead distribution is used because there is a greater motive head causing water circulation. This fact makes the use of smaller mains possible, resulting in some advantage in first cost, and in a more positive operation.

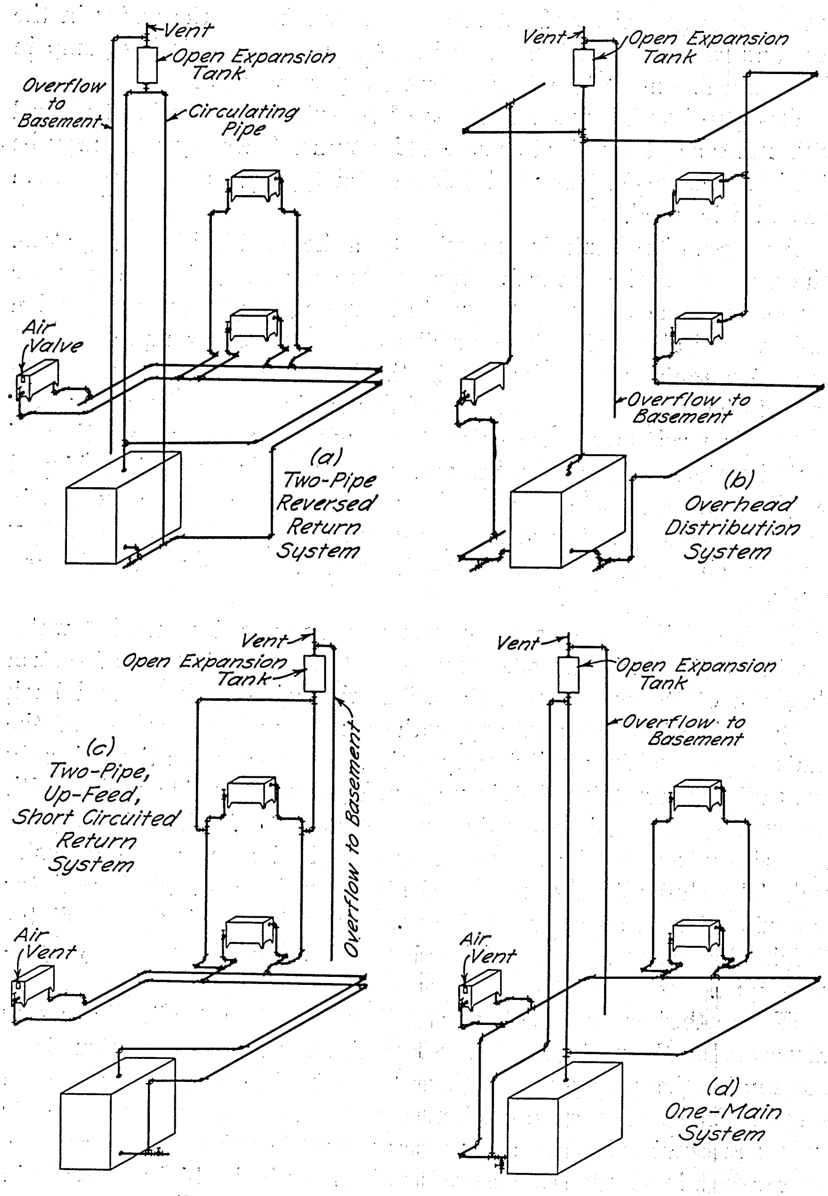

FIG. 4. TYPICAL HOT-WATER HEATING SYSTEMS

The two-pipe up-feed short-circuited return (sometimes called direct return) hot-water heating system is illustrated in Figure 4c. In this system, the length of the path of the water supplying each radiator is different. The radiator connections are similar to those used in types previously shown. The use of a second-floor radiator used to "pull" or provide more positive flow in a first-floor radiator is illustrated here. Because of the wide variation in the lengths of the paths of water supplying the radiators, great care must be used in balancing the resistance in the several circuits so that each radiator will be supplied with its just share of hot water. This balancing of resistance must be done largely by the selection of the proper pipe sizes. The limits imposed on pipe size selection by the commercial pipe sizes available often necessitate the use of orifice inserts to balance the flow properly. The orifices have the advantage in that they may be changed easily to overcome any initial unbalance in the system.

In the two-pipe hot-water heating systems just discussed, water flows through only one radiator during its circulation through the system. In the one-main (often called one-pipe) system, water may flow successively through a number of radiators, being supplied at lower temperatures to those radiators furthest removed from the boiler. These latter radiators must be proportionally larger in size. A one-main hot-water heating system is shown in Figure 4d. Essential air vents on the radiators are shown. The connections to the individual radiators are the usual two-pipe connections, both inlet and outlet connections being made to the single main. The first cost of this system is lower than that of the two-pipe systems, but more care must be exercised in the selection of radiator sizes for adequate and uniform heating.

The selection of proper pipe and radiator sizes, the grading of the pipes for quick venting, the covering of mains and returns with insulating material, and the equalization of the distribution of flow are important points to be checked in gravity hot water heating systems.

The forced hot water system may be patterned along the lines of any of the systems outlined for gravity flow with the one-main system finding increasing favor. The forced system has the advantages of positive flow, more frequent water circulation, and shorter heating-up time, and it lends itself to automatic temperature control. The addition of a circulating pump to a gravity system is frequently used as a means for correcting unsatisfactory gravity installations. This invariably aids the system but is not a cure-all.

TABLE 2
COMPARATIVE FEATURES OF HEATING SYSTEMS

Feature	Gravity-flow Warm Air	One-pipe Steam	Gravity-flow Hot Water
Relative cost for elementary system—lowest first	1	2	3
Efficiency in the use of fuel (same)	....	....	....
Ability to maintain required temperature (same)	....	....	....
"Dirty" when properly installed	No	No	No
"Dirty" due to faulty installation or lack of repair	Yes	No	No
Radiant heat available in room	No	Yes	Yes
Ability to use any kind of fuel (same)	....	....	....
Susceptible to temperature control	Yes	Yes	Yes
Relative "heating up" time—lowest first	1	2	3
Relative cooling time—lowest first	1	2	3
Occupies living space	No	Yes†	Yes†
Intereferes with use of floor space	Yes	Yes†	Yes†
Intereferes with use of basement space	Maximum	Minimum	Minimum
Adaptable to any type of building	Limited to 50 ft. x 50 ft. not more than 3 floors.	Yes	Yes
Can the distributing system be adapted for use in performing other air conditioning functions?			
Filter	Yes	No	No
Air Motion	Yes	No	No
De-odorizing (no satisfactory equipment available)	...	...	...
Humidification	Yes	Yes	No
De-humidification	Yes	No*	No*
Cooling	Yes	No*	No*

*Except with drips on pipes and radiators.
†Except when recessed convectors are used.

Two systems, not now widely used, deserve mention. Electric heating by electric radiant heaters, or by concealed convector units with fans, is available, and is used where the electric rates are attractive. Panel heating, a system of radiant heating, is accomplished by hot-water or steam-heated pipes buried in walls or ceilings just behind the plaster. As a consequence, the entire wall radiates heat, enabling the occupants of the room to feel comfortable at comparatively low room-air temperatures. This system has met with some use in England but has not been viewed favorably in this country.

After having viewed a skeleton of each of the commonly used types and systems the home owner often asks: "Which system is the best?" The answer is: "The one which serves your purposes best." Each system has its advantages and limitations. Table 2 has been prepared as a means of making some of the more obvious comparisons quickly. It has the fault of any table of the sort in that some of the qualifying remarks must be left to be inferred for the sake of compactness.

Examining the three types of systems most commonly used for residential heating according to their cost, recently published bids released by government agencies show that the gravity warm-air system is least expensive in first cost, the one-pipe steam system is next, and the gravity hot-water system is still more expensive. For any given house, such comparisons must be based on bids for that particular installation to be of any great value.

The efficiency, as expressed in fuel cost for a well-designed system, should be the same for each system, regardless of type, using the same fuel as a common basis for comparison.

Other comparisons made in the table are self-explanatory. There are always exceptions to any rule and the final guide in selection becomes a matter of the opinions and prejudices of the owner, and the relative emphasis placed on each factor by him. The table has been confined to these three systems because they represent common domestic usage. To include all of the various steam systems, forced-warm-air and forced-hot-water systems would introduce too many qualifying factors, and reduce the value of the comparison.

VIII. ESTIMATING THE HUMIDIFICATION REQUIREMENTS OF RESIDENCES

WILLIAM H. SEVERNS*

The atmosphere in which human beings exist consists of dry air and varying amounts of water vapor. The components of dry air and its possible impurities such as dust, bacteria, toxic gases, etc. will not be discussed here. The water vapor is steam which, when perfectly formed, is clear and colorless as is clear dry air.

Steam or water vapor may exist in either of two states, i.e., in a saturated, or in a superheated condition. Pure water may be vaporized at various temperatures depending upon the absolute pressure exerted upon it. The higher the pressure to which the water is subjected the higher its boiling temperature and conversely the lower the pressure the lower the vaporization temperature. Vapor which exists at a temperature corresponding to its absolute pressure is termed saturated and vapor which exists at a temperature greater than that corresponding to its absolute pressure is in a superheated state. The amount of water vapor that can be mixed with air is a variable quantity which depends upon the dry-bulb temperature of the air.

The dry-bulb temperature is the temperature of air as indicated by any thermometer having a moisture-free bulb which is not affected by water vapor held in the air. The wet-bulb temperature of air is the lowest temperature which a water wetted body will attain when exposed to the air when it is in motion. Both dry- and wet-bulb temperatures may be obtained by use of a psychrometer, one form of which is illustrated by Fig. 1.

This particular instrument is known as a sling psychrometer. It consists of two matched thermometers which are accurately calibrated and which are mounted upon a frame that may be whirled about the pivot handle. The bulb of the thermometer used to determine wet-bulb temperatures is covered with one layer of very thin gauze which should be moistened with distilled water when a reading is to be taken. When dry- and wet-bulb temperatures are to be determined the instrument is rotated rapidly in the air being investigated and the readings of both thermometers taken when the wet-bulb temperature is a minimum. Positive motion of the air must be induced over the wet-bulb thermometer; otherwise, its reading will be inaccurate. Ordinary hygrometers standing in still air often give misleading and inaccurate wet-bulb temperature indications which are generally too high. Ac-

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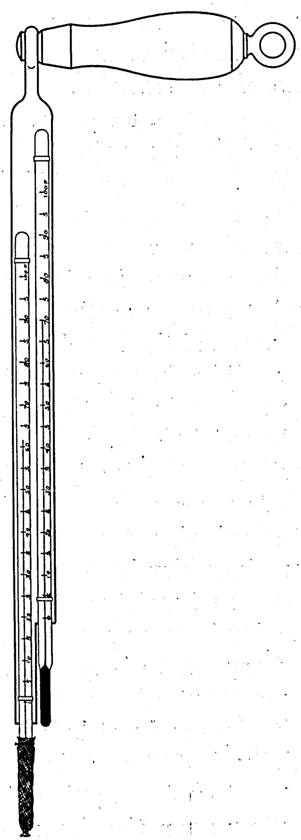

FIG. 1. SLING PSYCHROMETER

curate determinations of both dry- and wet-bulb temperatures are essential in psychrometric work as it is from these data that the moisture and total heat held by the air may be determined. The wet-bulb temperature is doubly important as it fixes the amount of heat held by a unit of air and water vapor at any dry-bulb temperature.

The depression of the wet-bulb temperature below that of the dry-bulb is produced by the evaporation of moisture at the wetted thermometer bulb. Sensible heat is transformed to latent heat when the evaporation takes place and the amount of evaporation that may occur at the moistened thermometer bulb is dependent upon the quantity of moisture mixed with the air. Sensible heat is that form of heat energy which, when added to or abstracted from a body, will produce a change of temperature of the body. Latent heat when added to or abstracted from a material will cause a change of the physical state of the material, either at melting or boiling but does not produce a temperature change. The unit of heat as used by heating and air conditioning engineers is the British thermal unit, B.t.u., which is 1/180 of the amount of heat necessary to raise the temperature of one pound of pure water from 32 to 212 deg. F. when the water is subjected to an atmospheric pressure of 14.7 lb. per sq. in. The greater the amount of moisture mixed with air maintained at a given dry-bulb temperature, the less will be the lowering of the wet-bulb temperature, and the less the amount of moisture held within the air the greater will be the amount of the wet-bulb depression.

The water vapor mixed with dry air is termed humidity. Absolute humidity is the weight of water vapor, in pounds or grains, per cubic foot of the mixture. Specific humidity of air is the weight of water vapor, in pounds or grains, per pound of dry air.

Air is saturated when the maximum amount of water vapor possible, at any given dry-bulb temperature, is mixed with it. The maximum amount of moisture which may be mixed with air is increased as the dry-bulb temperature is increased, and is decreased as the dry-bulb temperature is lowered. When air is saturated it holds the maximum amount of saturated vapor possible at its dry bulb temperature, its dry- and wet-bulb temperatures are identical, and its relative humidity is 100 per cent. When air is not saturated its wet-bulb temperature is less than its dry-bulb temperature by an amount depending upon the degree of saturation existent. Relative humidity is a ratio, usually expressed as a percentage, which indicates the degree of saturation existing in a space due to the water vapor present.

The moisture, in the form of vapor, held within an enclosure occupies the space as though the air were not present, and exerts its partial pressure, which, together with the partial pressure of the dry air, constitutes the total pressure of the mixture. When the air is saturated the partial pressure of the vapor is that of saturated steam at the same temperature and pressure. When the air is only partially saturated, that is when the relative humidity is less than 100 per cent, the vapor pressure is less than that of saturated steam corresponding to the dry-bulb temperature, and the vapor of the air is in a superheated condition.

The weight of water vapor required to saturate one pound of dry air is calculated in the following manner. The case of air at 71.5 deg.

F. and 29.92 in. of mercury atmospheric pressure is taken, as use will be made of the data later. From "Steam Tables and Mollier Diagram," by Keenan, it may be ascertained that the volume of one pound of saturated steam at 71.5 deg. F. is 828.2 cu. ft. and its vapor pressure is 0.382 lb. per sq. in. abs. The density of the steam is

$\frac{1}{828.2} = 0.001208$ lb. per cu. ft. The barometric pressure of 29.92 in.

of mercury represents a pressure of $29.92 \times 0.491 = 14.696$ lb. per sq. in., and the partial pressure of the dry air is $14.696 - 0.382 = 14.314$ lb. per sq. in. abs. The fundamental gas law equation is $PV = MRT$

or $V = \frac{MRT}{P}$, where V is the volume of gas in cu. ft., P is the absolute

pressure in lb. per sq. ft., M is the weight of gas in lb., R is a constant, 53.35 for air, and T is the absolute temperature of the gas in deg. F. In the case under consideration, M is taken as one pound,

and $V = \frac{1 \times 53.35\ (459.6 + 71.5)}{14.314 \times 144} = 13.74$ cu. ft., the volume of one

lb. of dry air. Hence, the moisture required to saturate one pound of dry air, for the conditions stated, is $13.74 \times 0.001208 = 0.01659$ lb.

Any reduction of the dry-bulb temperature of saturated air will cause condensation of some of its water vapor. When the air is not saturated, the dry-bulb temperature to which the air must be lowered, in order to produce water vapor condensation, is less than either the original dry-bulb or the original wet-bulb temperature. The dry- and wet-bulb temperature at which condensation of water vapor from air may be started is the dewpoint temperature. This temperature is dependent upon the absolute humidity of the air. If the absolute humidity is high, the dewpoint is higher than when the absolute humidity is low. Everyone is familiar with the collection, under certain conditions, of either a film of moisture or drops of water on the outside surface of a drinking glass containing either cold or ice water. Such an example illustrates the cooling of air and water vapor at the cold surface to produce vapor condensation and deposit. The air in contact with the cold surface has been cooled to or below its dewpoint temperature.

With the foregoing fundamental definitions in mind, the next considerations will be those of the humidification of residences. During the heating season the outside air temperatures vary considerably, and at times drop to rather low values. Consequently, the absolute humidities of the outside air fluctuate with outside air temperature changes. As the capacity of the air to carry moisture decreases with a decrease of its dry-bulb temperature, the amount of vapor held by the air is very small at temperatures of 0 deg. F. or less, even though its relative humidity is high, or even though the air is saturated under such conditions.

Humidification of the air during the heating season is desirable for the production of healthful and comfortable conditions, and for the protection of the interior finish of the structure, the furniture, the rugs, carpets, and other furnishings. Air deficient in moisture produces a dryness of the membranes of the nasal passages and throat, thereby promoting the occurrence of colds and respiratory ills. Dry air absorbs moisture from floors, doors, and furniture, causing shrinkage and the opening of cracks in the woodwork and the falling apart of pieces of furniture. The fibres of rugs and carpets become more brittle when extremely dry, thereby causing more rapid deterioration of such furnishings. Furthermore, the production of static charges of electricity is an annoyance to the householder in periods of cold weather when the air of the house is allowed to become low in moisture content.

No inhabited structure is absolutely air tight but permits the inleakage of outside air, termed infiltration, and the outleakage of a corresponding amount of warm inside air, which is known as exfiltration. These processes add materially to the heating load of the structure, as the outleaking air carries away heat, and the inleaking cold air must be warmed to the room temperature, otherwise, the rooms will not be comfortably heated. Infiltration and exfiltration also increase the amount of moisture which must be liberated within the space to maintain healthful and comfortable conditions. With low outside air temperatures, the absolute humidity of the inleaking air is low; consequently, when the air is warmed to room temperature, its relative humidity will be low unless moisture is added to it. The outgoing air takes moisture away with it so that, unless moisture is continuously added, both the absolute humidity and the relative humidity of the warmed space will not be sufficient.

The amount of air inleakage, and the corresponding outleakage of air from a structure, are dependent upon a number of factors. These items include: the height of the structure, the inside and outside air temperature conditions, the wind movement, the porosity and workmanship of the building walls, and the number and the width of

TABLE 1
AIR CHANGES PER HOUR DUE TO INFILTRATION

Room or Building	Number of Air Changes Taking Place per Hour
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance halls	2 to 3
Reception halls	2
Living rooms	1 to 2
Dining rooms	1 to 2
Bath rooms	2
Drug stores	2 to 3
Churches, factories, etc.	½ to 3

cracks in the walls and about window and door openings. The tightness of the building may be improved by the use of paint and wall paper on the interior surfaces of exterior walls, by caulking cracks about window and door frames, and by the use of either weather stripping or storm doors and storm sash at window and door openings. Both window stripping and storm sash when properly applied will do much toward reducing the hourly infiltration of air into the structure. Storm sash and storm doors have an advantage in that the heat transmission losses occurring at those areas where they are used are materially reduced below those which occur when such equipment is not used. When the heat losses, due to transmission through window glass and doors, are reduced the inside surface temperatures of such areas will be greater than they would be with a greater heat loss occurring with the same inside-outside air temperature differential. Inside surface temperatures are important in residence humidification problems as will be shown later.

Considerable investigative work has been done relative to the leakage of air through various types of walls, and also through the cracks about window and door openings. These test data are valuable if the engineer can modify them to suit the conditions of the structure with which he is concerned. The chief problems in the use of such data are the widths of and the lengths of the cracks which should be considered, as well as the probable wind velocity to be used. Another scheme of estimating the infiltration and the exfiltration of a structure is based on the estimated number of air changes occurring within the structure each hour. Recommended values as given in the 1936 edition of the A.S.H. and V.E. Guide are shown in Table 1.

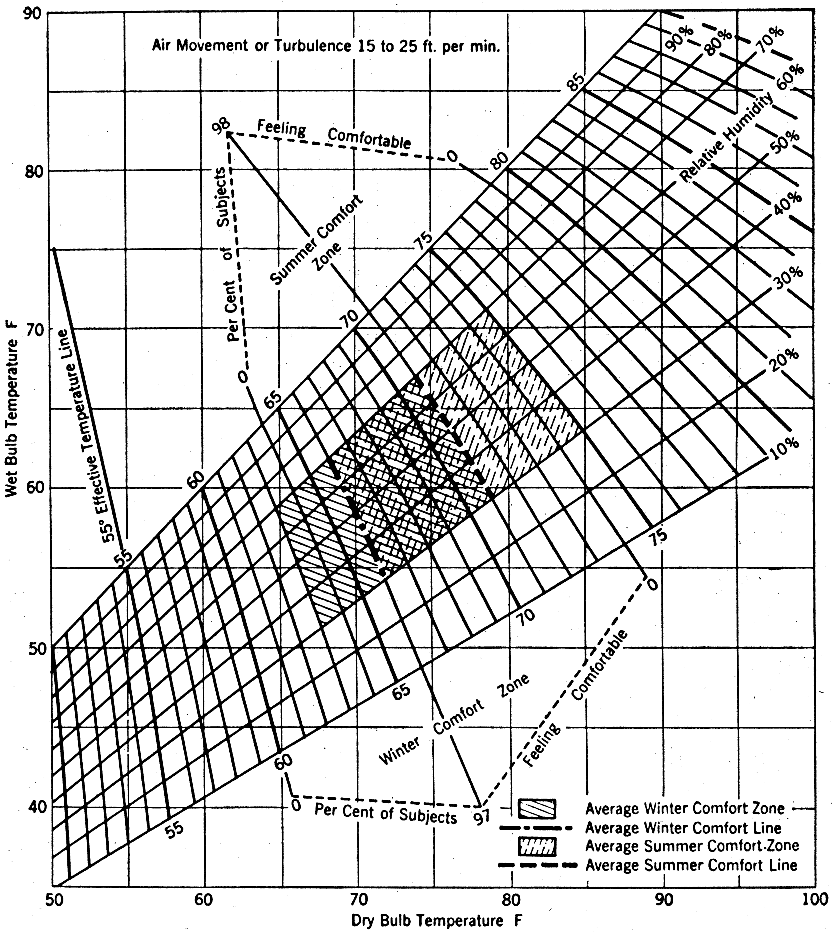

FIG. 2. COMFORT OR EFFECTIVE TEMPERATURE CHART FOR AIR VELOCITIES OF 15 TO 25 F.P.M. (STILL AIR)

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Under no condition should the number of air changes per hour be taken as less than one-half for any building. Estimates of the amount of air leaking into and out of a structure per hour, based on the number of air changes, are just as accurate as the assumed number of changes, as all other factors entering into the calculation are definite. Thus, in the case of a residence having 13 800 cu. ft. of occupied space, if the structure is equipped with storm sash and doors properly fitted, the number of air changes per hour may be taken as from ¾ to 1. Under such conditions, using one air change, the volume of air leaking into and out of the structure is 13 800 $\times$ 1 = 13 800 cu. ft. per hour measured at room temperature and pressure. This amount of air will provide adequate ventilation for eight adults.

The amount of moisture to be maintained in the air of the residence depends upon the conditions deemed necessary for comfort, and also upon the physical limitations of the structure. Reference to the comfort chart, Fig. 2, indicates that an effective temperature of 66 deg. F. gives the optimum condition of comfort, in the winter, for most people. Reference to the comfort chart and psychrometric tables establishes the conditions of dry-bulb temperature, relative humidity, and dew-point temperature (see Table 2) which will produce an effective temperature of 66 deg. F. when the dry-bulb temperatures range from 70 to 74 deg. F. Some individuals are prone to recommend a relative humidity of 50 per cent or even higher without thought of the difficulties involved in the maintenance of such conditions.

The dewpoint temperatures as given in Table 2 indicate the temperatures above which the inside surfaces of exposed walls and glass areas must be kept if condensation is not to occur upon them. The accumulation of condensed vapor on glass areas in the form of frost and ice which will melt when the sunshine strikes the surface is not desirable. Damage and discoloration of the wood of window stools will result when water accumulates upon them. Due to the accumulation of dust upon them, glass surfaces generally appear dirty after they have been wet. Therefore, when humidification of air is accomplished in the winter time, provision should be made to maintain interior surfaces of exposed walls and glass at a temperature above that of the dewpoint of the air held within the structure.

The calculated overall coefficient of heat transmission, for a frame wall having boxing and weather boarding on the outside, lath and plaster on the inside, and without insulation, is 0.26 B.t.u. per hour, per sq. ft. of area, per deg. F. difference in temperature between that of the outside and the inside air. The theoretical inside surface temperatures for such a wall are as indicated by Table 3 for various inside and outside air temperatures.

The inside surface temperatures shown by Table 3 are all above the dewpoint temperatures of air for the conditions indicated by Table 2, and no condensation on the interior surfaces of the wall may be expected for any of the conditions enumerated in Table 3. An investigation of the theoretical inside surface temperatures of glass areas for the given inside and outside air temperatures is desirable. Both

TABLE 2
DEWPOINT TEMPERATURES FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES

Dry-bulb Temperature of Air deg. F.	Relative Humidity per cent	Dewpoint Temperature deg. F.	Dry-bulb Temperature of Air deg. F.	Relative Humidity per cent	Dewpoint Temperature deg. F.
70	50	50.8	72	30	39.3
70.5	45	48.5	73	20	30.4
71	40	45.9	73.8	15	24.8
71.5	35	42.9			

TABLE 3
CALCULATED INSIDE SURFACE TEMPERATURES OF A FRAME WALL FOR VARIOUS INSIDE AND OUTSIDE AIR TEMPERATURES

Inside Air Temperature deg. F.	Outside Air Temperature deg. F.	Inside Surface Temperature deg. F.	Inside Air Temperature deg. F.	Outside Air Temperature deg. F.	Inside Surface Temperature deg. F.
70	0	59.0	71.5	−10	58.7
70	−10	57.4	72	0	60.7
70.5	0	59.4	72	−10	59.1
70.5	−10	57.9	73	0	61.5
71	0	59.9	73	−10	60.0
71	−10	58.2	73.8	0	62.1
71.5	0	60.2	73.8	−10	60.6

TABLE 4
CALCULATED INSIDE SURFACE TEMPERATURES OF SINGLE AND DOUBLE GLASS WITH AN AIR SPACE FOR VARIOUS INSIDE AND OUTSIDE AIR TEMPERATURES

Inside Air Temperature deg. F.	Outside Air Temperature deg. F.	Inside Surface Temperatures deg. F.	
		Single Glass	Double Glass with an Air Space
70	0	17.2	49.0
70	−10	9.7	46.0
70.5	0	17.4	49.4
70.5	−10	9.9	46.4
71	0	17.5	49.7
71	−10	10.0	46.7
71.5	0	17.7	50.1
71.5	−10	10.1	47.1
72	0	17.7	50.4
72	−10	10.2	47.5
73	0	18.0	51.0
73	−10	10.5	48.1
73.8	0	18.2	51.7
73.8	−10	10.7	48.7

single thickness of glass and double glass panels with an air space between the two glass panes are considered in the derivation of the surface temperatures given by Table 4. The commonly-used coefficient of heat transmission for single glass is 1.13, and for double glass with an air space between the panes 0.45 B.t.u. per hr., per sq. ft., per deg. F.

The results of investigations dealing with inside surface temperatures of window glass under winter conditions have been published recently by Emswiler and Randall in the April 1936 issue of the Journal Section of the A.S.H. and V.E. which appears in "Heating, Piping and Air Conditioning." These investigators established that with single glass the difference between the air temperature of the room and the temperature of the inside surface of the glass of windows is 67 per cent of the inside–outside air temperature difference. For windows fitted with ordinary storm sash the percentage is 30. When the data of Emswiler and Randall are used in calculating inside surface temperatures of glass, the values obtained for single glass are somewhat higher than those given in Table 4, and those obtained for double glass are the same as those in Table 4.

Calculations reveal that, with a single thickness of glass, condensation of moisture with its disagreeable results will be pronounced with an outside air temperature of 34 deg. F. or less when the relative humidity inside the space is 35 per cent, with a dry-bulb air temperature of 71.5 deg. F. Consequently, effective humidification cannot be said to be practical in cold weather when only a single thickness of glass is used in window and door openings. When the heat losses at window and door openings are reduced by the use of storm sash and storm doors effective humidification is practical when low outside air temperatures exist. If an inside air temperature of 71.5 deg. F. dry-bulb and a relative humidity of 35 per cent are taken as being feasible and satisfactory, without excessive condensation of water vapor except when the outside air temperature falls below —10 deg. F., a basis of estimating the amount of moisture for humidification purposes is established for a structure which is reasonably tight.

As an example of a calculation involving the humidification requirements the case of a residence having an occupied space of 13 800 cu. ft. and which has one air change per hour may be taken. The residence air is to be maintained at a dry-bulb temperature of 71.5 deg. F. and a relative humidity of 35 per cent when the outside air is saturated and has a dry-bulb temperature of —10 deg. F. The amount of moisture required for humidification purposes is to be found, and the extra heat required to effect the humidification is to be calculated.

The air infiltration is estimated to be 13 800 cu. ft. of air per hour, measured at 71.5 deg. F. and a barometric pressure of 29.92 in. of mercury, and the weight of air involved per hour is $13\ 800 \div 13.38 = 1031$ lb. The value 13.38 is the specific volume of one pound of dry air at 71.5 deg. F. and 29.92 in. of mercury. The specific humidity of air at 71.5 deg. F. and 35 per cent relative humidity is $0.01659 \times 0.35 = 0.005807$ lb. per lb. of dry air. The specific humidity of air at -10 deg. F. and 100 per cent relative humidity is 0.000459 lb. per lb. of dry air. Therefore, the moisture which must be added to the inleaking air per hour is $0.005807 - 0.000459 = 0.00535$ lb. per lb. of dry air, and the moisture requirements per hour are $1031 \times 0.00535 = 5.52$ lb. or $5.52 \div 8.34 = 0.661$ gallons. Assuming no other sources of moisture supply, the humidification requirements per 24 hours are $0.661 \times 24 = 15.9$ gallons of water. If the air leakage amounts to ¾ of a change per hour instead of one change, the foregoing value becomes $15.9 \times ¾ = 11.9$ gallons per 24 hours and the hourly evaporation of water must be 4.15 lb.

The calculated hourly heat losses for the residence are 885 B.t.u. per degree of inside–outside air temperature difference. If by reason of humidification the necessary dry-bulb temperature of the inside air can be reduced 3.5 deg. F. the hourly heat losses from the structure may be reduced by an amount equal to $885 \times 3.5 = 3100$ B.t.u. Humidification, however accomplished, requires the addition of heat to water to produce vapor. A simple method of calculating the heat added to produce the vapor is to consider that the water is made to boil under atmospheric pressure at 212 deg. F. The excess heat added to the water over that held by the vapor in its final condition is credited as sensible heat which is available for heating the house. This method gives results which check reasonably well with other methods of calculation, and can be applied to any condition, provided the proper numerical quantities for the weight of water to be evaporated, the inside air temperature, and the temperature of the water supply are inserted.

$$h = W_w\ [(212 - t_f) + 970.2 - 0.45\ (212 - t_r)]$$

h = heat necessary to effect humidification, B.t.u. per hr.

W_w = weight of water to be evaporated per hr., lb.

t_f = temperature of the supply water, deg. F.; 50 deg. F. in this case.

t_r = dry-bulb temperature of inside air, deg. F.; 71.5 deg. F. in this case.

Substituting in the equation,

$$h = 5.52\ [(212 - 50) + 970.2 - 0.45\ (212 - 71.5)]$$
$$= 5.52\ [162 + 970.2 - 63.5]$$
$$= 5.52 \times 1068.7$$
$$= 5900 \text{ B.t.u. per hour.}$$

The excess of heat required for humidification over that saved by humidification is $5900 - 3100 = 2800$ B.t.u. per hour. The additional hourly requirements of coal, having a heating value of 12 000 B.t.u. per lb. as fired, and used with an efficiency of 60 per cent, are $\frac{2800}{0.6 \times 12\ 000} = \frac{2800}{7200} = 0.39$ lb. The use of this small extra amount of fuel during cold weather can be justified by the more healthful conditions existent in the house, and the protection afforded to moisture-bearing objects within it.

IX. FACTORS AFFECTING FUEL SAVING

BY S. KONZO*

1. *Introduction.*—The amount of fuel required to heat a given structure is dependent on the nature of the construction of the building, the type of heating installation, and the method of operation of the heating plant. Improvements in building construction and modifications in the operation of the heating plant will affect to a greater or lesser extent the amount of fuel required. It is the purpose of this paper to discuss some of the improvements and modifications possible in the ordinary heating system, and to present approximate percentages of fuel savings that can be effected with each change. In addition to the results obtained in connection with the tests conducted in the Warm Air Research Residence, results from other investigations have been used.

If a structure is to be maintained at a temperature greater than that of the air outdoors, sufficient heat must be supplied to the structure to offset the heat loss from it to the outdoors. It is evident that for a given outdoor condition any change in the structure which will change the amount of this heat loss will affect the amount of heat to be supplied to the structure, which in turn will affect the amount of fuel required to produce this heat. In fact, the most effective means of making substantial reductions in fuel consumption consist in improvements in the house construction in the form of weatherstripping, insulation, and storm sash.

2. *Fuel Savings by Changes in House Construction.*—

(a) Storm Windows and Doors

It is obvious that the percentage of reduction in fuel consumption with and without the use of storm doors and windows is dependent on the nature of the wall construction and the ratio of exposed window surface to net wall surface. For a given room or house, storm windows and doors will effect a larger percentage saving when the wall is well heat-insulated than when it is not. Also, for two rooms of the same size having the same wall construction, but different ratios of window surface to net wall surface, the percentage of fuel saving will be greater when storm windows are applied to the room which has the larger ratio of exposed window surface to net wall surface, than it will be when the storm windows are applied to the room having the smaller ratio.

With due recognition of the fact that the fuel savings resulting

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from the use of storm windows and doors in any given installation may not be generally applicable to all installations, nevertheless it is of interest to note the results obtained from tests* conducted in the Research Residence.

The Residence is a three-story structure of standard frame construction. The wall section consists of weather boarding, building paper, sheathing, six-inch studding, wood lath, and plaster with rough sand finish. The walls are not insulated, and no weatherstripping is used at the windows and doors. The total space heated during these tests consisted of three rooms, a sun parlor, a breakfast nook, and a hall-way on the first story; three rooms, a bath room, and a hall-way on the second story; and two rooms, a bath room, and a hall-way on the third story. The total volume of this heated space, from which the basement was excluded, was approximately 17 540 cubic feet. The calculated heat loss was approximately 137 500 B.t.u. per hr. at an indoor–outdoor temperature difference of 70 deg. F.

The heating plant consisted of a coal-fired furnace used in connection with a forced-air heating system. The control of the heating plant was accomplished by means of a room thermostat operating to open and close the ash pit damper and to start and stop the fan. The room thermostat was located on an inside wall of the dining room at a height of 5 ft. from the floor, and was adjusted to maintain an air temperature of approximately 71 deg. F. at this level in all of the rooms of the Residence.

One series of tests was run with the Residence equipped with storm windows and door, and one series was run without such equipment. For the first-mentioned series all of the windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly-fitting storm sash. As shown in Fig. 1, felt stripping was placed along all four contact edges and the storm sash was clamped tightly to the window casing by means of screws. The areas of window and door openings, and of wall surfaces, and the ratios of openings to wall surfaces are given in Table I.

Data on the amount of fuel required to operate the Research Residence with and without storm windows and doors are presented in Fig. 2, in which the daily fuel consumption, in pounds of coal, is plotted against the difference in temperature between indoors and outdoors.

*A complete discussion on tests in the Research Residence is found in the paper entitled, "Fuel Savings Resulting from the Use of Storm Windows and Doors" by A. P. Kratz and S. Konzo, A.S.H. and V.E. Journal, Section of Heating, Piping and Air Conditioning, December, 1935.

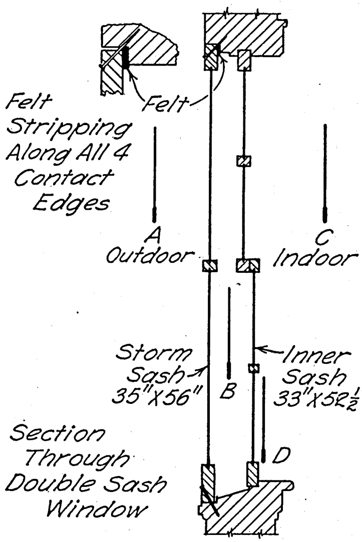


FIG. 1. METHOD OF INSTALLING STORM SASH

TABLE 1
DATA ON WINDOW AND WALL SURFACES

1	Number of window openings	50
2	Number of windows equipped with storm sash	48
3	Number of door openings to outdoors	2
4	Number of storm doors	1
5	Area of exposed window openings, sq. ft.	525
6	Area of windows equipped with storm sash, sq. ft.	522
7	Area of exposed door, sq. ft.	24.5
8	Area of double door, sq. ft.	24.5
9	Gross area of exposed structure, sq. ft.	3004.5
10	Net area of exposed wall (windows and doors excluded), sq. ft.	2455
11	Ratio of storm windows and doors to total exposed openings, per cent	99.4
12	Ratio of total exposed openings to gross wall, per cent	17.5
13	Ratio of total exposed openings to net wall, per cent	21.4

The curves show that the average daily amount of fuel required to heat the residence when the outdoor temperature was 40 deg. F., which corresponds closely to the mean seasonal temperature in Urbana, Illinois, was 100 lb. when storm doors and windows were not used, and 81 lb. when storm doors and windows were used. This represents a saving in fuel consumption of 19 per cent attributable to storm windows and doors. The saving in milder weather was somewhat less, but in severer weather increased up to a value of 21 per cent for zero weather, or at an indoor–outdoor temperature difference of 70 deg. F. The results, therefore, indicate that a saving of approximately 20 per cent in the seasonal fuel consumption could be

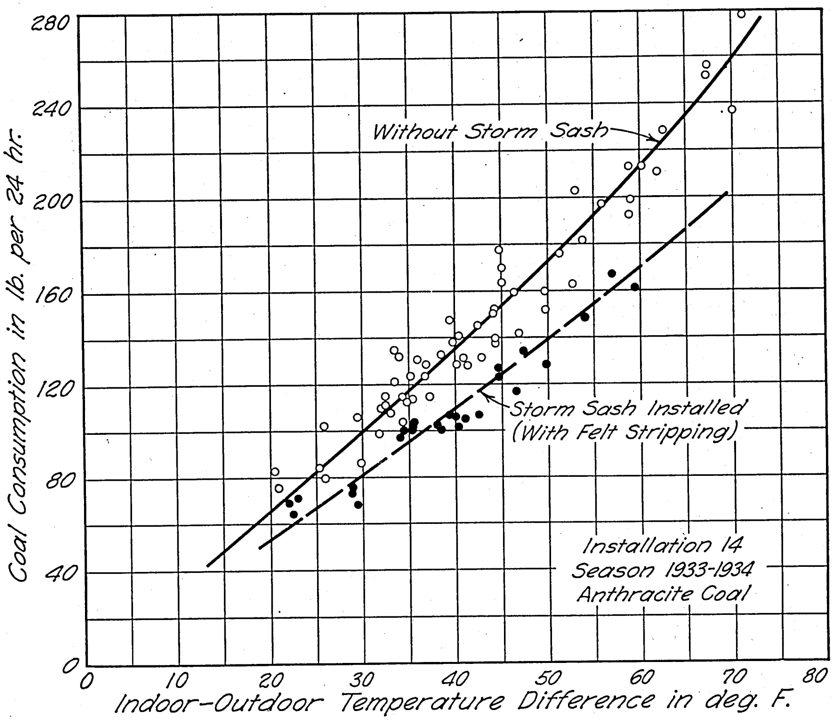


FIG. 2. FUEL CONSUMPTION WITH AND WITHOUT STORM SASH

reasonably attributed to the installation of storm doors and windows on the Research Residence.

The calculated reduction in heat losses was obtained by computing the heat losses from the structure both with and without storm windows and doors. The difference between the two calculated values would be accounted for in the items involving infiltration and heat transmission through windows and doors. These items were therefore calculated for a zero day, or for an indoor–outdoor temperature difference of 70 deg. F., and added to the basic heat loss of 73 770 B.t.u. per hr. which took place through walls, ceilings, floors, and all parts of the structure exclusive of the exposed windows and the front door. The data used for the calculation of the infiltration and heat transmission losses through the windows and door are given in Table 2, for which the coefficients were obtained from the American Society of Heating and Ventilating Engineers Guide for 1936.

From Table 2 it may be observed that the calculated heat loss was 137 570 B.t.u. per hr. for the building not equipped with storm windows

TABLE 2
CALCULATED HEAT LOSS DATA, BASED ON 70 DEG. F. TEMPERATURE DIFFERENCE

No.	Item	Without Storm Sash	With Storm Sash	With Weather Stripping
1	Heat loss through walls, floors, and ceilings, B.t.u. per hr.	73 770	73 770	73 770
2	Lineal feet of crack around windows (one half of total)	356	356	356
3	Lineal feet of crack around door	21	21	21
4	Total area of windows, sq. ft.	525	525	525
5	Area of windows with storm sash, sq. ft. (with weather stripping)	......	522	522
6	Area of windows without storm sash, (without weather stripping)	525	3	3
7	Area of door, sq. ft.	24.5	24.5	24.5
8	Infiltration coefficient for windows, B.t.u. per lineal ft. of crack per deg. F. per hr.	0.74*	0.34*	0.43†
9	Infiltration coefficient for door, B.t.u. per lineal ft. of crack per deg. F. per hr.	2.00	1.00	1.00
10	Coefficient of heat transmission for windows, B.t.u. per sq. ft. per deg. F. per hr.	1.13	0.45	1.13
11	Coefficient of heat transmission for door, B.t.u. per sq. ft. per deg. F. per hr.	0.52	0.52	0.52
12	Calculated heat loss through doors and windows, B.t.u. per hr.	63 800	27 070	54 606
13	Total calculated heat loss from building, B.t.u. per hr.	137 570	100 840	128 376
14	Calculated saving, per cent	......	26.7	6.7
15	Actual saving from tests, per cent	......	21.0	......

*Based on a well fitted window having 5/64-in. crack and 1/32-in. clearance.
†Based on infiltration of 23.6 cu. ft. per lineal ft. of crack. See A.S.H. and V.E. Guide of 1936, p. 135, Table 2, under column for 15 mile per hour wind velocity.

and door, and 100 840 B.t.u. per hr. for the building equipped with storm windows and door. This represented a computed saving of 26.7 per cent as compared with the actual saving of 21 per cent shown at an indoor–outdoor temperature difference of 70 deg. F. by the test curves in Fig. 2. This may be regarded as fairly close agreement, considering the uncertainties which necessarily accompany the computation of infiltration losses. It is possible that all fuel savings based on calculated values of maximum heat transmission may tend to be higher than the actual fuel savings obtained under average weather conditions. The calculated values in Sections (b) and (c) to follow should, therefore, be regarded as maximum theoretical savings.

In addition to fuel saving, other advantages incident to the operation with storm sash became evident. The tightly-fitting storm sash practically eliminated the entrance of soot, which in the case of the unprotected windows sifted in and collected in sufficient quantity on the white window stools to make daily cleaning necessary. The use of storm windows also enabled the maintenance of higher indoor relative humidities without condensation on the windows. They were also effective in reducing the downward draft of cold air usually present with unprotected windows. The immediate effect of this reduction in draft was an increase in the air temperature in the living zone of the room, particularly near the floor. This increase in the air temperature in the living zone, together with the increase in the surface temperature of the glass surfaces exposed to the occupant, resulted in an increase in the comfort conditions in the room when the same indoor temperature was maintained at the breathing level. Also, with a fixed setting of the controlling thermostat there was a marked reduction in the total time of fan operation in the forced-air system, accompanying a reduction in fuel consumption, when the storm windows were installed.

The advantages cited as resulting from the use of storm windows emphasize the importance of making adequate provision to reduce the heat losses at what may be regarded as the most vulnerable part of the structure from the standpoint of the heating installation, namely, the doors and windows. The problem of heating a room becomes greatly simplified when the ordinary unprotected windows are replaced by windows adequately protected by either weatherstripping, double-glass, or tightly fitted storm sash, although weatherstripping or double-glass alone are not as effective as tightly fitting storm windows.

(b) Calculated Savings due to Weatherstripping

From Table 2, columns 1 and 3, it may be observed that the calculated heat loss was 137 570 B.t.u. per hr. for the building not equipped with storm windows or weatherstripping, and 128 376 B.t.u. per hr. for the building equipped with weatherstripping on all windows and the exposed door. This represents a computed saving of 6.7 per cent. These items were calculated for a zero day, and for a wind velocity of 15 miles per hour. For average weather conditions in Urbana with wind velocities much less than 15 miles per hr. the fuel savings will probably be less than the percentages stated. If the same degree of correlation were obtained between the calculated saving and actual saving as was obtained in the case of the storm windows, the probable fuel saving with all the windows equipped with weatherstripping would be of the order of 5.3 per cent.

In addition to this saving in the cost of fuel, weatherstripping should also add very largely to the comfort of the occupants by reducing drafts created by excessive air leakage.

It is obvious, as in the case of storm windows, that the percentage of reduction in fuel consumption with and without the use of weatherstripping is dependent on the nature of the wall construction, the tightness of the windows, and the amount of linear feet of crackage around the windows and doors in the house. Furthermore, for a given room or house, the calculated percentage of fuel saving will be larger when weatherstripping is applied to loosely-fitting windows than when it is applied to tightly-fitting windows and doors.

(c) Calculated Savings with Insulation

Certain details of construction in the Research Residence, where the space directly under the roof has been utilized for dormitory space, make it rather difficult to calculate the possible fuel savings with the use of insulation. It is more convenient for the present purpose to consider the probable fuel saving that would be obtained in a typical two-story house, with unfloored attic space. For this comparison the data presented by C. G. Segeler* giving the relative amounts of heat loss from various parts of a house have been used. These values, which were an average obtained from an analysis of 200 homes, showed that the proportions of total heat loss from the various parts of the house were as follows: 16.2 per cent through the roof, 27.0 per cent through the walls, 25.8 per cent through the glass, 24.6 per cent by infiltration, 4.3 per cent through the door, and 2.1 per cent through other sources, including the floor.

For these calculations it was assumed that the total heat loss for the "typical" house during maximum weather conditions was 100 000 B.t.u. per hour, composed of a roof loss of 16 200 B.t.u., a wall loss of 27 000 B.t.u., a glass loss of 25 800 B.t.u., an infiltration loss of 24 600 B.t.u., a door loss of 4300 B.t.u., and a floor loss of 2100 B.t.u. per hr. (see item (a) in Table 3). In addition it was assumed that the wall section was composed of yellow pine lap siding, building paper, 1-in. fir sheathing, 3⅝ in. studding space, and wood lath and plaster, and that the ceiling adjacent to the attic space was composed of wood lath and plaster attached to unfloored joists. The over-all heat transmission coefficients for the uninsulated wall and ceiling were respectively 0.245 and 0.62 B.t.u. per sq. ft. per deg. F. difference in temperature. The addition of 1-in., 2-in., 3-in., and 3⅝-in. thicknesses of mineral wool insulation having a thermal conductivity of 0.27 B.t.u. per sq. ft. per deg. F. per inch of thickness, reduced the values of the over-all coefficient of heat transmission of the wall from the original value of 0.245 B.t.u. to values of 0.129 B.t.u., 0.087 B.t.u., 0.066 B.t.u., and 0.060 B.t.u., respectively. Similarly the addition of 1-in., 2-in., 3-in., and 3⅝-in. thicknesses of mineral wool reduced the

*Discussion by C. C. Segeler in A.S.H. and V.E. Transactions of 1928, vol. 34, p. 473. The percentages of heat loss were average values obtained from an analysis of 200 homes heated with gas fuel.

values of the over-all coefficient of the ceiling from the original value of 0.62 B.t.u. to values of 0.188 B.t.u., 0.111 B.t.u., 0.079 B.t.u., and 0.066 B.t.u., respectively.

The heat losses through the wall and the ceiling were reduced by amounts proportional to the reduction of the heat transmission coefficients, and the revised figures for the two losses were used in obtaining the total heat loss from the house for the various conditions listed in Table 3.

It may be noted from the values listed in Table 3, items (b) and (e), that although the addition of the first inch of insulation to the original wall reduced the total heat loss from the house by approximately 12.8 per cent, the addition of 2⅝ in. more insulation reduced the total heat loss by only another 7.5 per cent. In the case of the wall insulation, the maximum reduction in heat loss amounting to 20.3 per cent (e) was obtained when the studding space was completely filled with insulation. Similarly in the case of the ceiling insulation, the maximum reduction in heat loss amounting to 14.5 per cent (i) was obtained when the space between the joists was completely filled with insulation. When a layer of insulation 3⅝ in. thick was added to both the sidewalls and ceilings, the total reduction in heat loss from the house amounted to 34.8 per cent (j).

The reductions in heat loss obtained with the use of weather-stripping on all the windows and doors, and with the use of storm windows and storm doors, are listed as items (k) and (l) respectively. The maximum reduction in the heat loss from the structure, amounting to 66.1 per cent (m) was obtained when both the sidewalls and ceilings were insulated with a 3⅝ in. thickness of insulation, and the house was completely equipped with storm windows and storm doors.

It is probable that these calculated reductions in heat loss, which were based on maximum load conditions, may not exactly correspond to the fuel saving obtainable under ordinary weather conditions. The calculated values should only be regarded as maximum savings obtainable under conditions of full load. On the other hand, the calculated values for reduction of heat loss take no account of the change in comfort conditions resulting from the increases in temperature of the wall surfaces exposed to the occupant, the increase in air temperature in the living zone, and the reduction of drafts. Neither do they take into account the increase in comfort conditions in the summer resulting from the reduced house temperature.

It should be recognized that the return on the investment, whether that investment consist of storm sash, insulation, or any other fuel-

TABLE 3
CALCULATED HEAT LOSSES FOR TYPICAL RESIDENCE

Cases	Calculated Heat Loss Through Various Parts of House, in B.t.u. per hour							Reduction in Heat Loss per cent
	Roof	Walls	Glass	Infiltration	Door	Floor	Total	
(a) No insulation or storm sash*	16 200	27 000	25 800	24 600	4 300	2 100	100 000	0
(b) 1-in. insulation in sidewall†	16 200	14 200	25 800	24 600	4 300	2 100	87 200	12.8
(c) 2-in. insulation in sidewall	16 200	9 590	25 800	24 600	4 300	2 100	82 590	17.4
(d) 3-in. insulation in sidewall	16 200	7 290	25 800	24 600	4 300	2 100	80 290	19.7
(e) 3⅝-in. insulation in sidewall	16 200	6 620	25 800	24 600	4 300	2 100	79 640	20.3
(f) 1-in. insulation in ceiling‡	4 910	27 000	25 800	24 600	4 300	2 100	88 710	11.3
(g) 2-in. insulation in ceiling	2 900	27 000	25 800	24 600	4 300	2 100	86 700	13.3
(h) 3-in. insulation in ceiling	2 060	27 000	25 800	24 600	4 300	2 100	85 860	14.1
(i) 3⅝-in. insulation in ceiling	1 720	27 000	25 800	24 600	4 300	2 100	85 520	14.5
(j) 3⅝-in. insulation in walls and ceiling	1 720	6 620	25 800	24 600	4 300	2 100	65 140	34.8
(k) Weather stripping on all windows¶	16 200	27 000	25 800	14 760	4 300	2 100	90 160	9.8
(l) Storm windows and doors all over§	16 200	27 000	10 270	11 320	1 840	2 100	68 730	31.3
(m) 3⅝-in. insulation in walls and ceiling and storm windows and door	1 720	6 620	10 270	11 320	1 840	2 100	33 870	66.1

*Values from discussion by C. G. Segeler in A.S.H. and V.E. Transactions of 1928, Vol. 34, p. 473; average values from 200 homes.

†Original wall assumed to be of ordinary frame construction consisting of yellow pine lap siding, building paper, 1-in. fir sheathing, 3⅝-in. studding space, wood lath and plaster, over-all coefficient = 0.245. With addition of wool insulation having conductivity of 0.27 B.t.u. per inch, the overall coefficient of heat transmission becomes: 0.129 B.t.u. for 1 in. insulation, 0.087 for 2 in., 0.066 for 3 in., and 0.060 for 3⅝ in.

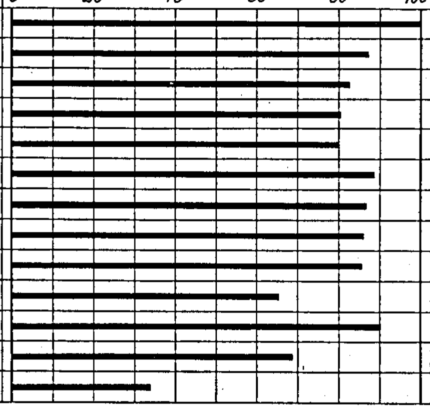
‡Original ceiling assumed to consist of wood lath and plaster, unfloored, over-all coefficient of 0.62. With addition of wool insulation having conductivity of 0.27 B.t.u. per inch, the over-all coefficient of heat transmission becomes: 0.188 B.t.u. for 1 in. insulation, 0.111 B.t.u. for 2 in., 0.079 B.t.u. for 3 in., and 0.066 B.t.u. for 3⅝ in. fill.

¶Leakage without weatherstripping = 39.3 cu. ft. per ft. of crack, with weatherstripping = 23.6 cu. ft. per ft. of crack.

§Over-all coefficient single window = 1.13 B.t.u., double window = 0.45 B.t.u. Infiltration values, single window = 0.74 B.t.u., storm windows = 0.34 B.t.u.

Calculated Heat Losses for Typical Residence

Case	Insulation		Weather Strip	Storm Sash	Calculated Hourly Heat Loss in Per Cent of Maximum Rate (0, 20, 40, 60, 80, 100)	Reduction, %
	Wall	Ceiling				
a	None	None	None	None		0
b	1-in.	None	None	None		12.8
c	2-in.	None	None	None		17.4
d	3-in.	None	None	None		19.7
e	3⅝-in.	None	None	None		20.3
f	None	1-in.	None	None		11.3
g	None	2-in.	None	None		13.3
h	None	3-in.	None	None		14.1
i	None	3⅝-in.	None	None		14.5
j	3⅝-in.	3⅝-in.	None	None		34.8
k	None	None	Yes	None		9.8
l	None	None	None	Yes		31.3
m	3⅝-in.	3⅝-in.	None	Yes		66.1



saving device, depends not only on such factors as the cost of fuel, the cost of the heating equipment, and the severity of the weather, but also on whether the conservation measure is planned before the house is built or is adopted at a later date. In fact the greatest returns on the investment are obtained when the conservation measures are planned before the house is built, and the cost of the investment is partially offset by the decrease in cost of the heating equipment for the house.

The greatest reductions in fuel cost can be obtained by substantial improvements in the house structure, and such improvements should be given careful consideration in existing as well as in proposed structures.

3. *Fuel Savings Effected by Differences in Plant Installation.*—

(a) Conversion Gas and Oil Burning Furnaces

When a furnace or boiler, which is designed for burning solid fuel, is converted to a unit for burning gaseous fuel, the thermal efficiency obtainable with the conversion unit may be almost as high as that obtainable with a furnace or boiler especially designed for gas combustion. This was demonstrated by the tests made by Professor R. B. Leckie* at Purdue University. Therefore, in the case of gaseous fuel

*See Purdue University Engineering Bulletin Research Series No. 36. "Tests of Gas Home-Heating Equipment" by R. B. Leckie and C. H. B. Hotchkiss, Vol. XV, No. 3, May 1931, also Research Series No. 48, "Investigation of the Use of Gaseous Fuel in Warm-Air Furnaces" by R. B. Leckie, Vol. XVIII, No. 6, November 1934.

very little gain in fuel economy is to be expected by replacing a properly-installed conversion unit with a unit specially designed for burning gas.

On the other hand, in the case of oil conversion units decided differences in fuel economy were obtained when a conversion unit was replaced in the Research Residence with a furnace especially designed for oil combustion. One series of tests was run with a gun type oil burner placed in a cast-iron, circular radiator type furnace having a total heating surface of approximately 60.3 sq. ft. The tests were made with the burner adjusted to burn oil at a rate of 13.0 lb. per hr., with the air inlet to the burner adjusted to maintain 9.5 per cent CO_2 in the flue gases, and with the circulating fan in the forced-air system adjusted to deliver approximately 1675 cu. ft. per minute. The second series of tests was run with the same oil burner placed in a steel furnace designed for oil combusion, having a total heating surface of approximately 102 sq. ft. The tests were made under identical conditions of operation, and data were secured over a wide range of outdoor weather conditions. The data on the amount of fuel required to operate the Research Residence with the two units are shown in Fig. 3 in which the daily fuel consumption, in pounds of oil, is plotted against the difference in temperature between the indoors and outdoors. The curves show that the average daily amount of fuel required to heat the Residence when the outdoor temperature was 40 deg. F., which corresponds closely to the mean seasonal temperature in Urbana, Illinois, was 82.5 lb. when the conversion unit was in use, and 73.0 lb. when the designed unit was in use. This represents a difference in fuel consumption of 11.5 per cent attributable to the difference in furnace design and construction.

In this connection similar data are desirable on the relative fuel consumption of a hand-fired coal furnace and the same furnace equipped with a mechanical coal stoker. It should be noted that the difference between the cost of heating a house with the use of a mechanical stoker and that of heating it with the use of a hand-fired coal plant is attributable not so much to the increase in thermal efficiency of the furnace or boiler as to the decrease in unit cost of the coal. In fact, any cost comparisons between hand-fired and stoker-fired coal plants are subject to extremely wide variations depending upon the cost of the fuel used as a basis of comparison. The use of a stoker will have the additional advantages of convenience, smokeless combustion of high-volatile coals, and more uniform residence temperatures.

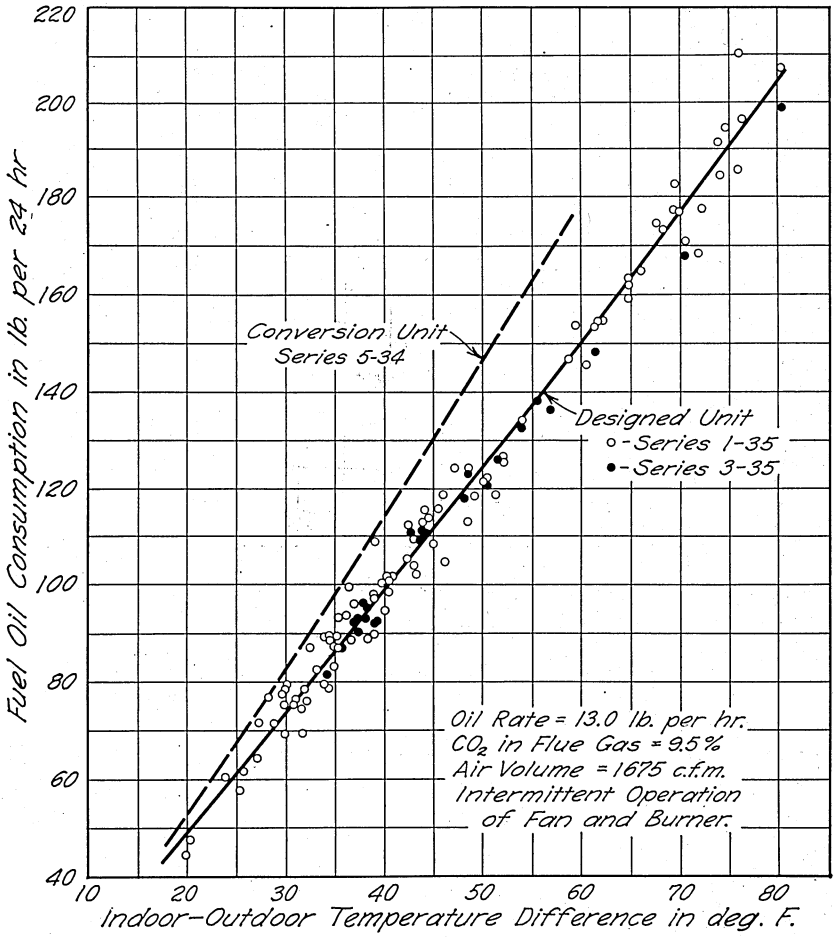


FIG. 3. FUEL CONSUMPTION CURVES FOR TWO OIL FURNACES

(b) Design Characteristics of Furnaces

Tests made in the Laboratory Plant at the University of Illinois* on four different types of warm-air furnaces indicated the importance of heating surface as a factor in determining the capacity of a furnace under conditions of equal combustion rate. The curves in Fig. 4 indicate that the relation between the ratio of heating surface to grate surface and unit capacity, for the types of furnaces tested, and over the range of ratios of heating surface to grate surface included by the

*"Investigation of Warm-Air Furnaces and Heating Systems, Part III," Univ. of Ill. Eng. Exp. Sta. Bul. 188, pp. 38-44.

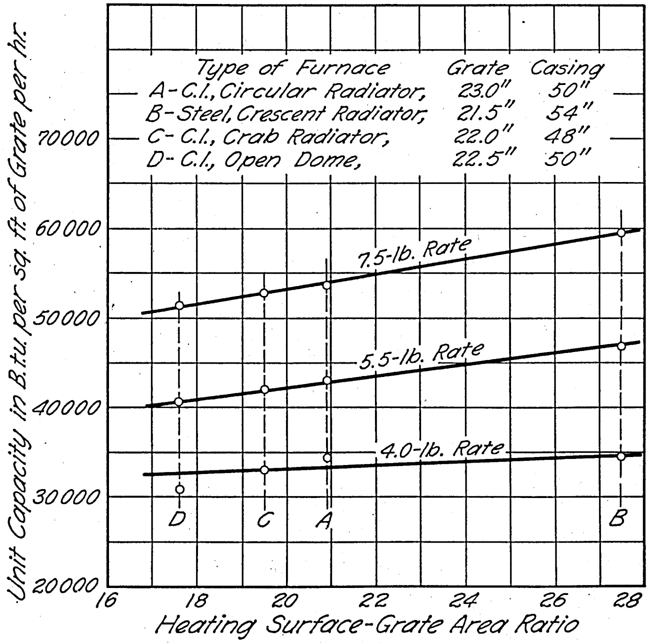


FIG. 4. EFFECT OF HEATING SURFACE AREA ON UNIT CAPACITY

furnaces tested, can best be represented by straight lines, particularly at medium and high combustion rates. If a combustion rate of 7.5-lb. is regarded as typical, and a ratio of heating surface to grate surface of 20 to 1 is taken as a base, the slope of the line is such that an increase of approximately 2 per cent occurs in the capacity per sq. ft. of grate surface per hour for each unit of increase in the ratio of heating surface to grate surface. Under normal operating conditions in a residence a slight increase in the capacity would not be reflected in a corresponding increase in fuel saving. However, the results do indicate that, from the standpoint of thermal efficiency, the material used in the construction of the furnace is relatively unimportant, whereas, the relative amount of heating surface per unit area of grate is of some significance.

4. *Fuel Savings Effected by Differences in Plant Operation.*—

(a) Proper Condition of Combustion

The amount of heat wasted from the chimney in any heating installation is an irretrievable loss. This loss may be accounted for in

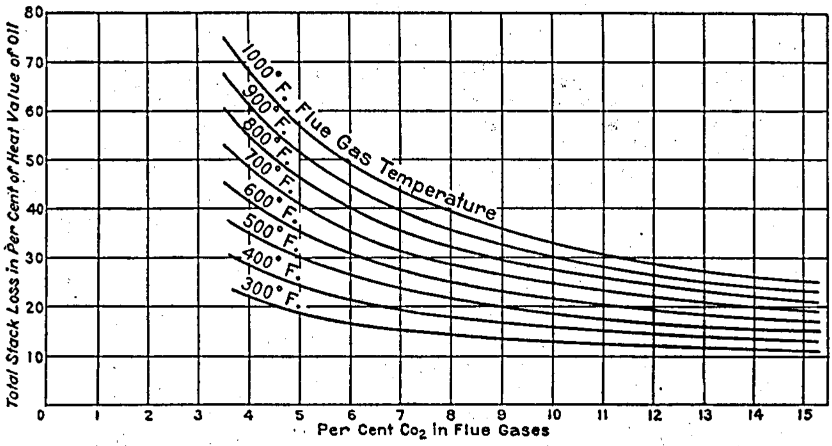


FIG. 5. STACK LOSS CURVES FOR COMBUSTION OF DOMESTIC OIL-BURNER DISTILLATES
(From Technical Bulletin No. 109, U. S. Department of Agriculture)

the unburned combustibles, in both solid and gaseous form, that result from incomplete combustion, in the uncondensed moisture in the flue gas, and in the sensible heat remaining in the hot gases.

The unburned combustibles in the flue gas may be composed of solids in the form of carbon or higher hydro-carbons, and gases in the form of carbon monoxide, hydrogen, or hydro-carbons. Material improvement, particularly in hand-fired coal plants, may be effected by proper firing methods, which minimize the formation of free hydrogen and carbon monoxide. The loss represented by carbon in the smoke may be small, but it may indicate the presence of CO and hydrogen.

Even under conditions of complete combustion, however, a substantial amount of heat is lost through the chimney, the magnitude of which is dependent on the temperature and composition of the flue gas. In the case of oil combustion, for instance, these stack losses can be computed for a given flue gas temperature and for a given percentage of CO_2 existing in the flue gas. Figure 5* shows typical stack-loss curves applicable to combustion of domestic oil-burner distillates. Similar curves can be derived for gaseous or solid fuels when the chemical constituency of the fuels is known.

It may be noted in Fig. 5 that, for a given percentage of CO_2 in the flue gases, the stack losses increase as the temperature of the flue gases increase. An oil-burning furnace having a large amount of heat-

*Curves reproduced from U. S. Dept. of Agriculture Bulletin No. 109, "A Study of the Oil Burner as Applied to Domestic Heating" by Arthur H. Senner, July, 1929, Page 71, Fig. 33.

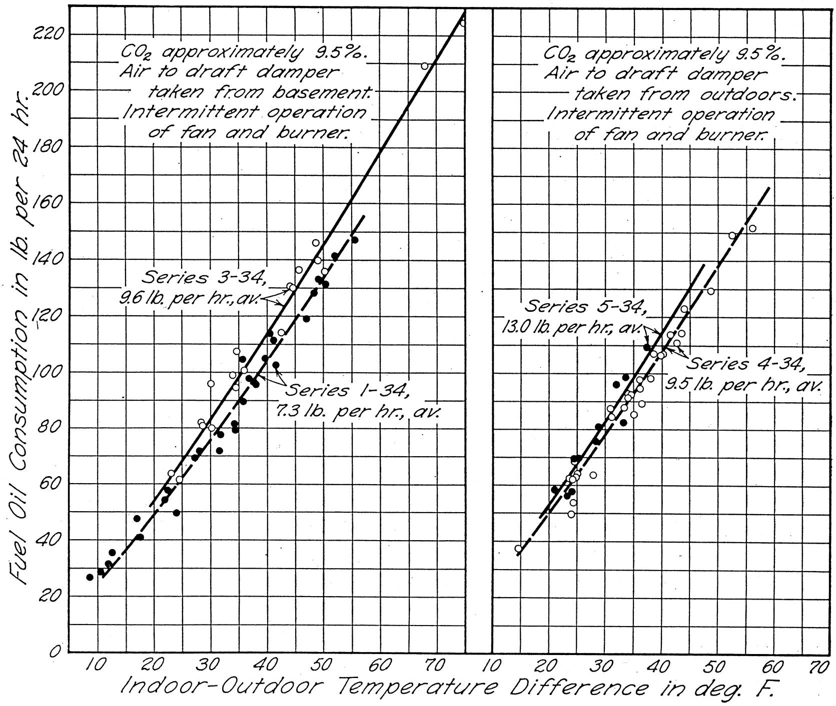


FIG. 6. DAILY FUEL CONSUMPTION CURVES FOR VARIOUS RATES OF OIL INPUT INTO CONVERSION OIL FURNACE WITH FORCED-AIR HEATING SYSTEM

ing surface will abstract a greater amount of heat from the hot gases, resulting in a lower flue gas temperature, than a furnace having a smaller amount of heating surface. The stack losses for the furnace having the large amount of heating surface will therefore be less than those for the furnace having the smaller amount of heating surface. This was found to be the case in the tests made in the Research Residence which were discussed in Section 3.

Also, for a given oil-burning furnace, operated to maintain a given percentage of CO_2 in the flue gas, the flue gas temperature will be directly dependent upon the rate of oil input to the furnace. As a result, as indicated by the curves in Fig. 6, the thermal efficiency of the furnace tends to decrease as the rate of oil combustion is increased.

It may also be observed from Fig. 5 that, for a given flue gas temperature, the stack losses will be diminished if the percentage of CO_2 in the flue gases is increased. In the case of oil combustion the

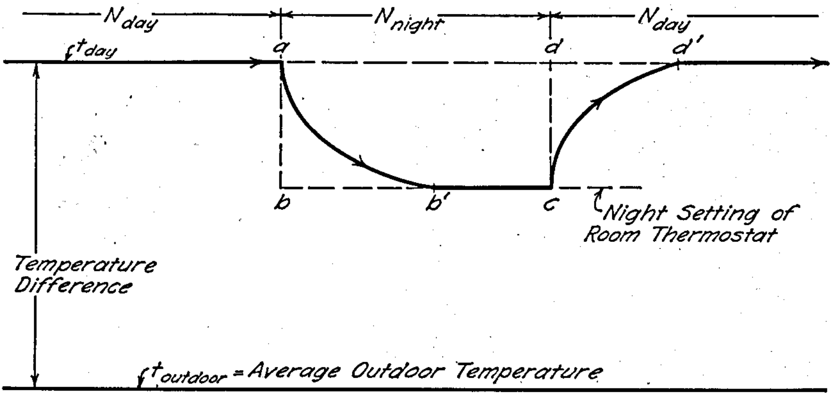


FIG. 7. ACTUAL CHANGES IN INDOOR TEMPERATURE AS INFLUENCED BY DELAY OR "LAG" IN COOLING AND WARMING OF STRUCTURE

air adjustment on the burner should be set to provide for the minimum amount of excess air that will not be accompanied by the presence of CO, smoke, or other operating difficulties. It may be noted that if the flue gas temperatures can be maintained at a value less than 500 deg. F. there is not much to be gained by attempting to increase the CO_2 composition in the flue gas beyond approximately 10 per cent.

(b) Soot Formation on Heating Surfaces

The accumulation of soot on the heating surfaces of a boiler or furnace tends to decrease the heat transmission from the hot flue gases to the circulating medium. The tests conducted at the Bureau of Mines* by Nichols and Augustine have indicated that the decrease in heat transmission and efficiency caused by the soot formation on heating surfaces was smaller than ordinarily believed to be the case, and much smaller than the decrease in thermal conductivity of soot alone.

The following excerpts are reprinted from the conclusions stated in the Bureau of Mines Report:

(1) The tests showed that in round, cast-iron boilers with three intermediate sections,—that is, a ratio of effective secondary to total heating surface of about 0.4—operated at the average rate for a heating season the insulating effect of deposits of soot decreased the heat absorbed by the boilers 2 to 7 per cent of the heat made available by the fuel, depending on the thickness of the deposit.

(2) The losses, expressed as a percentage of the coal burned, will be less and will be approximately proportional to the over-all efficiency. Thus, with

*See paper entitled, "Effect of Soot on Heat Transmission in Small Boilers," by P. N. Nicholls and C. E. Augustine. U. S. Bureau of Mines Report of Investigations 3272, Feb., 1935.

60 per cent efficiency, about 1¼ to 4¼ per cent of the coal will be wasted because of the insulating effect of the soot.

. . . .

(5) These values do not include the waste of fuel that may result from incomplete combustion due to insufficient draft caused by the soot clogging the passages or flue, which may, and usually will be, much greater than the loss caused by the insulating effect.

(c) Reduction of House Temperatures at Night

The reduction of the house temperatures during the night period, by means of manual or automatic adjustment of the setting of the room thermostat, tends to reduce the temperature difference maintained between indoors and outdoors and hence tends to reduce the fuel consumption required for a given outdoor weather condition. In the lower part of Fig. 7 is shown a line diagram which represents the temperature conditions maintained in the house with reduced setting of the room thermostat at night. The difference in heat loss from the structure when the house temperature is reduced at night as compared with the heat loss from the structure when the house is maintained at a constant temperature during the entire 24-hour period is represented diagrammatically in Fig. 7 by the area enclosed within *ab′ cd′ a*. It may be observed that the fuel savings effected by reducing the house temperatures at night will be dependent on the length of time during which the temperature is reduced, the temperature maintained during the night and the day, the rate of cooling of the house structure, the rate of rise of the house temperatures during the morning, the difference in efficiency of the combustion process during the periods of continuous and intermittent operation of the burner, and the outdoor temperature. The large number of variable factors make it very difficult to calculate accurately the probable fuel savings obtainable in any given installation.

Actual values of fuel savings were obtained from two series of tests conducted in the Research Residence. In the first series of tests made with a gas-fired, gravity warm-air furnace system, a decrease in gas consumption of approximately 4 to 8 per cent was effected. In the second series of tests made with an oil-fired, forced-air heating system a decrease in oil consumption of approximately 6 to 9 per cent was effected when the thermostat setting was reduced from 71 deg. F. to 60 deg. during the period between 10 p.m. and 4:30 a.m.

In case the house temperatures are not brought up rapidly in the morning, the reduction of temperatures at night will be accompanied by a sacrifice of comfort in the house during the morning hours, brought about by the influence of cold walls and floors persisting over most of this period. The heating plant must be capable of assuming the extra load during the warming-up period without damage to the equipment resulting from the long period of continuous operation. The possible economy in fuel consumption effected by reducing the house temperature to 60 deg. F. at night would probably be less in the case of coal-fired plants than in those of gas- or oil-fired plants. However, other factors might make it advisable to bank the coal fire at night and maintain a reduced rate of combustion in the furnace in order to prevent all of the fuel from burning out before morning.

5. *Conclusion.*—The most effective means of making substantial reductions in fuel consumption consist in improvements in the house construction in the form of weatherstripping, insulation, and storm sash. The problems of heating are likewise greatly simplified when such improvements are made.

In some cases, as in oil combustion, the fuel consumption will be less when the fuel is burned in a unit especially designed for that fuel. In all cases, those heating units that provide the greatest amount of effective heating surface will be the most efficient in transferring the heat from the flue gases to the circulating medium.

The stack losses from the chimney may be reduced to a minimum by maintaining clean heating surfaces, by providing the least amount of air sufficient for complete combustion, and by adjusting the combustion rate to a minimum value that will satisfy the heating requirements. In the case of gas-fired and oil-fired units, decreasing the house temperatures at night will effect a small decrease in fuel consumption.

X. CALCULATION OF THE REFRIGERATING LOAD

HORACE J. MACINTIRE*

The refrigerating engineer has been using for a quarter of a century the terms "manufactured weather" and "let every day be a good day" as applied to air conditioning in industry. The more recent application of refrigeration to comfort cooling in the home or in the office has modified the details, but has in no way altered the objective of the work. We manufacture a definite set of conditions involving temperature, humidity, and intensity of air motion, we control the direction of the air movement, the amount of dust the air contains, the maintenance of the air below a detectable concentration of odor, vapor, and smoke, and we even eliminate sound vibrations beyond a certain intensity level. We try to make the conditions of rest or work comfortable and every day a good day.

In the early days of comfort cooling, the popular advertisement was "30 deg. F. cooler inside," or later we attempted to secure conditions of 57 deg. F. dewpoint temperature. Both of these objectives were ill advised, as 30 deg. F. colder inside was not comfortable, and attempts to secure 57 deg. dewpoint temperature led to chaos. At the present time we feel that any effective temperature from 70 or 71 to 74 or 75 deg. F. will be comfortable to the average person within a short time after exposure.

However, the securing of comfortable conditions involves the lowering of the temperature within the enclosure and a reduction of the moisture content of the air. This requires the use of a machine, usually, which has the ability to reduce the air temperature to a dewpoint temperature sufficiently low so that the proper water vapor content of the air may be secured, and which has the required cooling capacity to not only condense out the surplus water vapor, but to absorb the sensible heat entering the office or residence.

1. *Factors Involved in Comfort Cooling.*—In comfort cooling, the sources of sensible and latent heat loads on the refrigerating equipment are as follows:

(a) The sensible heat *conducted* through the walls, windows, ceiling and roof.

(b) The *radiant* heat from the sun entering through the walls, roof and windows.

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*Professor of Refrigeration, University of Illinois.

(c) The sensible and latent heat brought into the building by uncontrolled *infiltration* or on account of controlled fresh air.

(d) The sensible and latent heat liberated by the *people present.*

(e) The *heat generated* by various machines, and appliances, such as electric lights, machines, and heating appliances.

2. *Conduction.*—If the temperatures inside and outside the home or office are t_i and t_o, respectively, the amount of heat leakage, Q_c, in B.t.u. per hour may be found by the expression:

$$Q_c = A_w U_w \, (t_o t_i) \text{ B.t.u. per hr.,}$$

where A_w is the area of the wall, window, ceiling, or partition, and U_w is the overall coefficient of heat transfer in B.t.u. per sq. ft. per deg. F. per hr.

The outside-air design temperature should be selected so that the maximum temperature over a period of years will be included during about 90 per cent of the cooling season. The space above the ceiling may be excessively hot, being as much as 120 to 130 deg. F. in the latitude of Urbana, Illinois.

3. *Solar Radiation.*—Solar radiation has to be handled differently from conduction. The maximum intensity of the sun varies with the latitude and the time of day, and in Urbana it may be as much as 210 for vertical walls, 280 for horizontal roofs, and 190 for vertical windows. The amount of heat (Q_r) absorbed by a roof or wall and transmitted to the interior may be given by*

$$Q_r = A_w \, F \, e \cdot I \text{ B.t.u. per hr.,}$$

where I = the actual solar radiation striking the wall surface.

e = the decimal part of the incident solar radiation absorbed by the wall; this varies from 0.4 for very light-colored surfaces like white paint to 0.9 for very dark surfaces like tar roofing material.

F = the decimal part of the absorbed solar radiation transferred into the inside of the wall = 0.23 U_w.

A_w = area of the wall in sq. ft.

Frequently an approximation may be made by adding, say, 25 deg. F. to the outside air temperature for that part of the outside wall exposed to the sun's rays.

4. *Sun Effect Through Windows.*—Although certain special glasses are capable of reducing the effect of the solar radiation through glass on the refrigeration load by from 16 to 24 per cent, yet usually from 90 to 95 per cent actually passes through the glass undiminished. The

*Faust, Levine and Urban, Journal Section, A.S.H. and V.E., August, 1935.

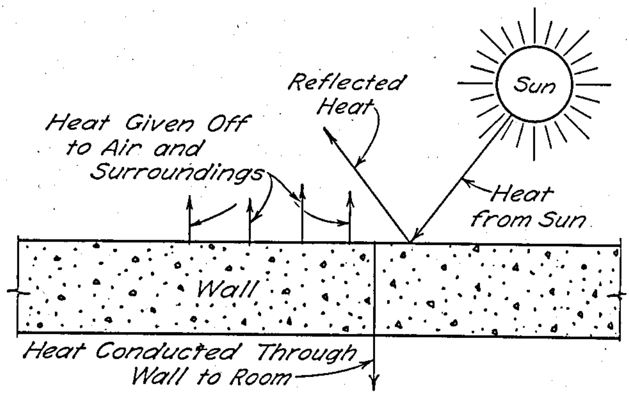

FIG. 1. HEAT FLOW THROUGH WALL OF RESIDENCE

amount of solar heat (Q_s) transmitted directly through the glass in B.t.u. per sq. ft. per hr. is given by the expression:

$$Q_s = A_g R_g,$$

where A_g = the area of the glass in sq. ft.

R_g = the amount of solar heat transmitted directly through the glass for the proper latitude and time of day.

Values for R_g must be obtained from suitable handbooks.

Solar radiation through the windows frequently accounts for a very large proportion of the refrigerating load. In a Detroit office building it amounted to 75 per cent, and in the Research Residence the heat gain through the windows alone was about 25 per cent. Some reduction of the refrigerating load may be obtained by nullifying the window losses. An outside awning or outside Venetian blind will reduce the solar radiation leakage by from 75 to 85 per cent, whereas an inside Venetian blind, or a buff colored shade, full drawn, will eliminate about 50 per cent.

5. *Infiltration.*—Most residences are constructed in such a manner as will permit an entire volume change of the air in the building with all doors and windows closed in less than one hour. The usual number of air changes per hour for detached houses is from 1½ to 2. In office buildings of recent construction the number of air changes in an office room was found by test to vary from 0.86 with doors and windows closed to 6.2 with the office door open 45 deg. If fresh air is admitted it should be controlled, and it should be filtered and dehumidified before entrance into the enclosure. That the effect of infiltration is a heavy one may be seen in referring to tests on the Research

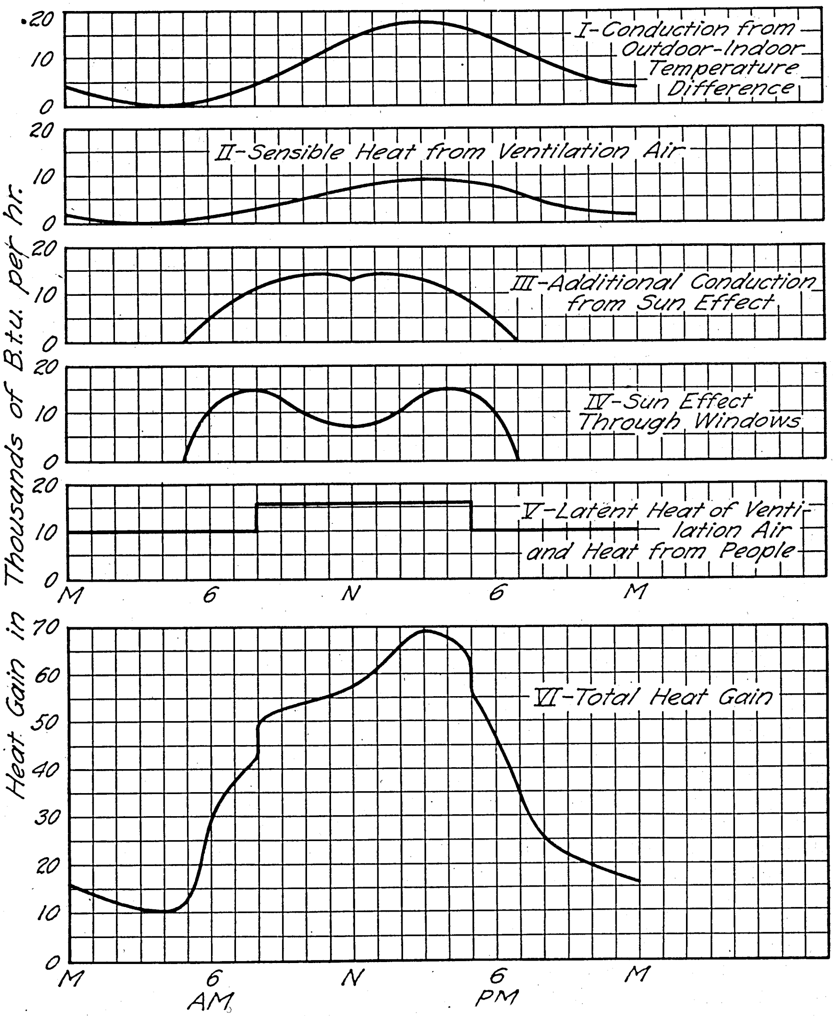

FIG. 2. TOTAL HEAT GAIN OR MAXIMUM LOAD ON REFRIGERATING MACHINE

Residence during the summer of 1932, when over 110 lb. of water vapor per 24 hr. was condensed from the air during the dehumidification, the greater part of which was assigned to the air of infiltration. Such an amount of condensed vapor would require 0.4 tons, or 800 lb. of ice—an amount that is excessive in a residence for the normally sized family. The refrigeration load due to controlled and uncontrolled fresh air must be calculated by use of psychrometric tables or diagrams.

6. *The Sensible and Latent Heat of the People Present.*—The heat dissipated by the average person is practically constant at 400 B.t.u. per hr. per person over a range of temperature from 65 to 90 deg. F. effective temperature. This is divided into sensible and latent heat, the sensible heat loss being a maximum at the lower air temperatures, and a minimum at about 98 deg F.

7. *Heat Generated by Various Machines and Appliances.*—The load on the refrigerating machine is about 3.4 B.t.u. per hr. per watt in the case of electric lights, toasters, electric irons, electric stoves, and motors. Gas has a calorific value of from 600 to 800 B.t.u. per cu. ft. Unless separately vented, most of this heat becomes a load on the refrigerating machine.

8. *The Maximum Load.*—The maximum load, from which the capacity of the refrigerating machine is determined, can be found only by adding the separate items with due respect to the time the load occurs. This may be done best by plotting as shown in Fig. 2.

9. *Conclusion.*—In an auditorium, restaurant, or other place for public gathering the cooling load may be principally a human load. In the residence or office, the cooling load probably is one due to solar radiation, heat leakage, and infiltration. From the point of view of heating as well as cooling, it seems that attempts to reduce these factors by storm windows and doors, weatherstripping, insulation in the walls and ceilings, and the use of some device to reduce the effect of solar radiation is a very good investment. Undoubtedly, the use of night air can be depended on to freshen the indoors satisfactorily, but after sunrise during the day time all doors and windows should be closed.

XI. RESEARCH IN SUMMER COOLING AT THE UNIVERSITY OF ILLINOIS

By M. K. Fahnestock*

The purpose of this paper is to give a brief résumé of the research work in summer cooling or summer air conditioning which has been done at the University of Illinois. The work starting in 1932 has been limited entirely to residential cooling studies carried on in the Research Residence,† using ice, mechanical refrigeration, water from the city mains, and outdoor air at night. The different methods of cooling, accompanied with the wide variations in weather experienced during the past four cooling seasons, have resulted in the accumulation of a large amount of data, of which only the most significant and practical can be included in this paper. The tests have been reported in detail in research papers‡ of the American Society of Heating and Ventilating Engineers and will ultimately comprise bulletins¶ of the Engineering Experiment Station. All of the data and figures appearing in this paper have been taken from these publications.

The tests were conducted by the Engineering Experiment Station of the University of Illinois in coöperation with the American Society of Heating and Ventilating Engineers and the National Warm Air Heating and Air Conditioning Association. They were made in the Department of Mechanical Engineering under the direction of Professor A. P. Kratz. Special acknowledgments are also due the several manufacturers who made financial contributions and loaned equipment for the work.

*Research Assistant Professor of Mechanical Engineering, University of Illinois.

†The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the National Warm Air Heating and Air Conditioning Association in December, 1924.

‡"Study of Summer Cooling in the Research Residence at the University of Illinois," by A. P. Kratz and S. Konzo. Transactions A.S.H. and V.E., Vol. 39, 1933, pp. 95-118.

"Study of Summer Cooling in the Research Residence for the Summer of 1933," by A. P. Kratz and S. Konzo. Transactions A.S.H. and V.E., Vol. 40, 1934, pp. 167-198.

"Study of Unit Room Coolers in the Research Residence," by A. P. Kratz, M. K. Fahnestock, and S. Konzo. A.S.H. and V.E. Journal Section of Heating, Piping and Air Conditioning, November, 1934, p. 483.

"Study of Summer Cooling in the Research Residence for the Summer of 1934," by A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick. A.S.H. and V.E. Journal Section of Heating, Piping and Air Conditioning, January, 1935, pp. 29-30.

"Study of Summer Cooling in the Research Residence Using Water from the City Water Mains," by A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick. A.S.H. and V.E. Journal Section of Heating, Piping and Air Conditioning, May 1936.

¶"Investigation of Summer Cooling in the Research Residence," University of Illinois Engineering Experiment Station Bulletin. (The manuscript for this bulletin is now being prepared and it will contain the results of the investigation when cooling with ice, mechanical refrigeration, and night air.)

FIG. 1. RESEARCH RESDENCE, WITH AWNINGS

1. *Description of Research Residence and Cooling Equipment.*—

The Research Residence

The Research Residence, shown in Fig. 1, is of a common type of frame construction. The wall section consists of bevel siding, building paper, sheathing, studding, wood lath and plaster. The roof is of copper shingles, blackened from soot and corrosion after several years' exposure to the elements. The walls are not insulated, and no weather-stripping, double-glazing, or storm sash is used at the windows or doors. The building faces to the south, and, with the exception of a short time during the first season's cooling studies, it has been equipped with awnings on all east, south, and west windows.

There are three stories in the residence, but for the cooling studies the third story was considered as an attic space, and was closed off by means of a door at the top of the stairs. Likewise, on the first story, the sun room was isolated by closing the door between it and the adjoining room. The total space cooled, in addition to six average

Leader Ducts			Stacks or Throats		Registers
No.	Name	Size	Floor	Size	Size
1	N.W. Bedr'm.	8x8½	2nd	5½x13	10x12
2	Kitchen	7x8	1st	7x14	12x14
3	N. Trunk, 1	8x12¾			
5	Bath	3x10	2nd	3x10	8x10
6	Bath	8x10			
7	S.W. Bedr'm.	8¾x5¾	2nd	3½x12	9x12
8	N. Trunk, 2	8¾x12			
9	E. Bedr'm.	6x12	2nd	6x12	10x12
10	Main, N. Tr.	8x32			
11	Main, S. Tr.	8x32			
13	Liv. Rm.	5½x13	1st	5½x13	10x12
14	Hall	7½x14	1st	7½x14	12x14
15	S. Trunk	8½x14			
17	Din. Rm	7x14			
18	Din. Rm.	5½x13	1st	5½x13	10x12

●-Represents Location of Squeeze Damper.

NOTE: Heating system ducts were "blocked off" at 6 places as shown. Ducts were not otherwise disturbed in any way.

Return Ducts			Grilles*
No.	Name	Size	Size
21	Hall-1	9x12½	6x40
22	Hall-2	9x13¾	6x40

*Wood Grilles (56% Free Area)

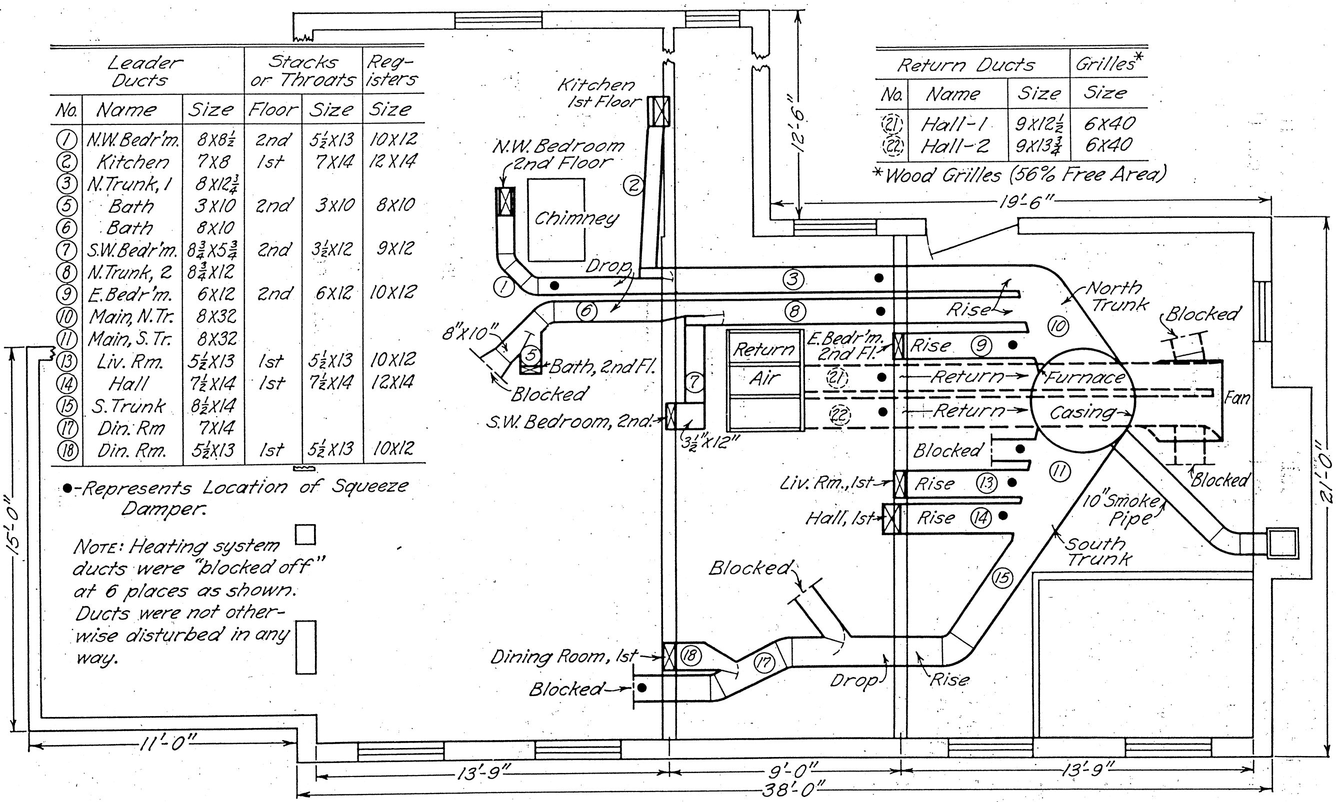

FIG. 2. BASEMENT PLAN SHOWING DUCT LAYOUT FOR FORCED-AIR SYSTEM IN USE FOR DISTRIBUTING COOL AIR

size rooms, included a breakfast room, a bath room, and a stairway, with interconnecting halls, making in all about 14 170 cu. ft. of space, of which 7300 cu. ft. were on the first story and 6870 cu. ft. were on the second story. The method of calculating the design cooling load has been revised from time to time, making use of more complete and accurate data as it has become available. The major revisions have been in relation to the load due to solar radiation and the effect of various window appurtenances. The design load for the space cooled in the residence calculated by the method outlined in the A.S.H. and V.E. Guide, 1934, was 37 500 B.t.u. per hour. In accordance with the recommendations in the Guide, an outdoor design temperature of 91 deg. F. was selected, and 80 deg. F. was used as the inside temperature. On the average there were about four persons in the house, and as there was practically no cooking, there was no heat from that source.

The Duct Layout

The basement plan and the forced-air heating plant are shown in Fig. 2. With a few minor changes the forced-air duct system used for heating was adapted for distributing cool air for the cooling studies. All return ducts, with the exception of the central one leading from the foot of the stairs on the first story, and which contained the cooling coil, were blocked. The delivery ducts to the third story and the sun room were blocked, as indicated, and by means of dampers in the ducts leading to the first and second stories the cooling loads on these two stories were balanced. The air velocities in the ducts did not exceed 750 ft. per min., and in most cases they were considerably below that value. The cool air was introduced into the various rooms through a common type of baseboard register, except in one case where a wall type of register located 7 ft. above the floor was used.

The Central Cooling Plants

Three different cooling plants, using ice, mechanical refrigeration, and water from the city mains, have been installed and operated in the Residence during the course of the cooling studies. Although the plants were assemblies of pieces of commercial apparatus, in most instances, additions, refinements, and operating conditions had to be modified from what would have been considered good commercial practice in order to facilitate the taking of test data. This was particularly true in the case of the plant using ice for cooling. In general, the arrangements of all of the plants were similar to that of the mechanical refrigeration plant shown in Fig. 3. With each installation

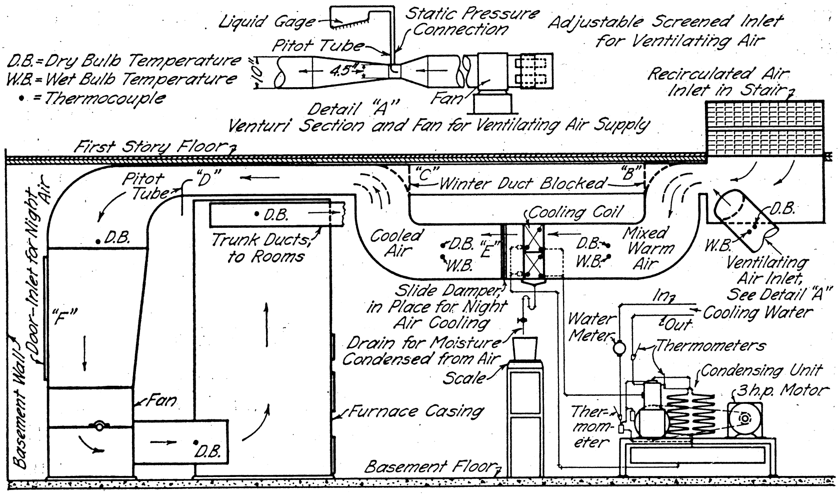

Fig. 3. Diagram of Cooling Plant with Mechanical Refrigeration

an extended surface cooling coil was located in a by-pass in the one central return duct and the flow of the refrigerant or cooling medium was always counter to the flow of the air. This arrangement provided that the coldest or leaving air came in contact with the tubes containing the coldest refrigerant. For the purpose of heating, the dampers at *B* and *C,* in Fig. 3, were adjusted and the air was delivered directly to the fan without passing through the cooling coil. The wet- and dry-bulb temperatures of the air as it entered and left the coils were taken with thermometers and thermocouples, and facilities were provided for collecting and weighing the moisture removed from the air during the cooling process. The cooled and dehumidified air after leaving the cooling coil passed directly to the fan and then to the furnace casing, whence it was distributed through ducts to the various rooms on the first and second stories of the residence.

During all tests, when cooling with ice, mechanical refrigeration, or water from the city mains, all windows on the first and second stories remained closed. The windows in the attic remained open when cooling with mechanical refrigeration and water from the city main, but they were closed when the tests using ice were made.

Details of the Plant Using Ice for Cooling

In addition to the air ducts, cooling coil, and fan, the plant using ice for cooling included an insulated bunker or tank having an ice capacity of 4000 lbs. The chilled water was pumped from the bottom of the tank through the cooling coil, which was six rows deep in the direction of air flow. Spray-heads located in the top of the tank sprayed the water returning from the cooling coil over the ice, which was placed on racks, and both the circulating water and the meltage drained to the bottom of the tank. By means of a constant level drain the amount of ice melted during any period of time could be determined, and with minor corrections this ice meltage represented the cooling load. It was this comparatively simple and convenient method of determining the cooling load that led to the selection of ice as the cooling medium for the first cooling studies made in 1932.* With this plant the winter return duct was not tightly blocked at *B* and *C* as shown in Fig. 3, but instead, a modulating by-pass damper was located at *B* and the damper at *C* was omitted entirely. Whenever cooling was needed a constant quantity of cold water was circulated through the coil, and the control of the air temperature in the residence was obtained by a thermostat located in the dining room on the first story. This thermostat operated by the by-pass damper causing larger or smaller quantities of return air to pass through the coiling coil, depending upon the load requirements. The fan used for heating was also used for cooling, and under the conditions of operation it delivered 1475 cu. ft. of air per min. which was equivalent to 6.2 air recirculations per hour. No provision was made for bringing air in from the outside for ventilating purposes, or for cooling with outside air at night. For the majority of the tests the house was equipped with awnings on all east, south, and west windows, but for the purpose of determining the effect of awnings in decreasing the cooling load a few tests were made with the awnings removed.

Details of the Plant Using Mechanical Refrigeration

As previously stated, Fig. 3 shows the arrangement of the cooling plant with mechanical refrigeration. The condensing unit consisted of a double-pipe condenser and a four-cylinder Freon compressor with a rating of 30 000 B.t.u. per hour. The compressor was driven with a 3 hp. motor. Cooling and dehumidification were accomplished by the direct expansion of Freon refrigerant in an extended surface coil or evaporator placed in the by-pass in the one central return duct. This coil was 4 rows deep in the direction of air flow.

In the common refrigeration cycle the warm air, in passing through the cooling coil or evaporator, gives up heat to the refrigerant within the coil, causing it to boil and change from a liquid to a gaseous state.

*See footnote p. 123.

The gas, at a relatively low temperature and pressure, is returned to the compressor, compressed to a higher pressure and discharged into the condenser. During the compression process the temperature of the gas is increased to a level above that of the water in the condenser. Thus, in the condenser the hot gaseous refrigerant can give up heat to the condenser cooling water, and in doing so condenses or returns to the liquid state. It is a continuous and closed cycle in which the refrigerant absorbs heat from the air and conveys it to the condenser where it is transmitted to the condenser cooling water and discharged outside of the building.

With this plant the winter return duct was tightly blocked at *B* and *C*, as shown in Fig. 3, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the plant was operating. The amount of cooling was controlled by means of a room thermostat placed in the hall on the second story, which served to start and stop the refrigerating unit in accordance with the cooling load required to maintain constant room temperature. The fan was run continuously through both on and off periods of the refrigerating unit, and under the conditions of operation it delivered approximately 1300 cu. ft. of air per minute. The same fan was used during the previous cooling studies when ice was the cooling medium. For the purpose of ventilation during the periods when the plant was operating, outdoor air equivalent to one air change per hour was brought in through the duct shown as Detail A in Fig. 3. Thus, of the total of 5.5 air changes per hour delivered by the fan, 4.5 air changes were recirculation of the air in the house and one was ventilating air from outdoors. The fan motor was one-third horsepower.

For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass duct on the down-stream side of the cooling coil, and a door, *F*, was placed in the re-circulating duct near the fan inlet. When cooling with outdoor air at night the basement door and the door in the recirculating duct were opened, and the slide damper was closed. Under these conditions of operation the fan delivered 2250 cu. ft. of air per minute, which was equivalent to 9.5 air changes per hour.

Details of the Plant Using Water from the City Mains

In general, the arrangement of the cooling plant using water from the city supply mains was the same as that of the mechanical refrigerating plant shown in Fig. 3. The cooling coil, consisting of 8 rows of finned tubes in the direction of air flow, was placed in the by-pass in the one central return duct. The center winter return duct was tightly blocked at *B* and *C*, as shown in Fig. 3, and all of the air delivered to the fan passed through the coil when the plant was operating. The water from the city service main, available at a temperature of approximately 58 deg. F., passed through a calibrated water meter before entering the coil. The operation of the plant was controlled by the room thermostat located in the hall on the second story, which served to start and stop the flow of water through the coil by operating a motor-driven valve placed on the outlet water line from the coil. The valve was either open or closed and the water consumption rate was 360 gal. per hour.

The cooling coil installed for economical and practical operation with water at a temperature of 58 deg F. was necessarily deeper in the direction of air flow than the coils required during the previous cooling studies, when water chilled with ice to a temperature of 35 deg. F., and Freon operating at a suction pressure corresponding to 43 deg. F., were used. The deeper coil increased the air resistance in the circulating system, and a larger fan was required in order to maintain an air delivery of 1300 cu. ft. of air per minute. The size of the motor driving the fan was three-fourths horsepower. A quantity of outdoor air, equivalent to one air change per hour, was brought in during the periods of plant operation for ventilating purposes. This was the same amount as used with the mechanical refrigerating plant.

The arrangement for cooling with outdoor air at night was practically the same as that shown in Fig. 3 and previously described under the mechanical refrigerating plant. The quantity of air delivered under these conditions of operation was approximately 2300 cu. ft. of air per minute, which was equivalent to 9.7 air changes per hour.

The Attic Fan for Cooling with Outdoor Air at Night

In addition to the facilities provided for cooling with outdoor air at night by means of the fan in the main air circulating system, during one cooling season, an attic fan shown in Fig. 4 was installed in the doorway at the head of the stairs leading from the second to the third story. This 24-inch propeller-type fan delivered air from the first and second stories into the third story or attic space whence it escaped through the windows on all sides. The use of a special duct and damper arrangement on the suction side of the fan, as shown in Fig. 4, permitted outdoor air to be drawn into the open first and second story windows at night, or to be drawn through a duct from a third story window in order to provide positive ventilation for the

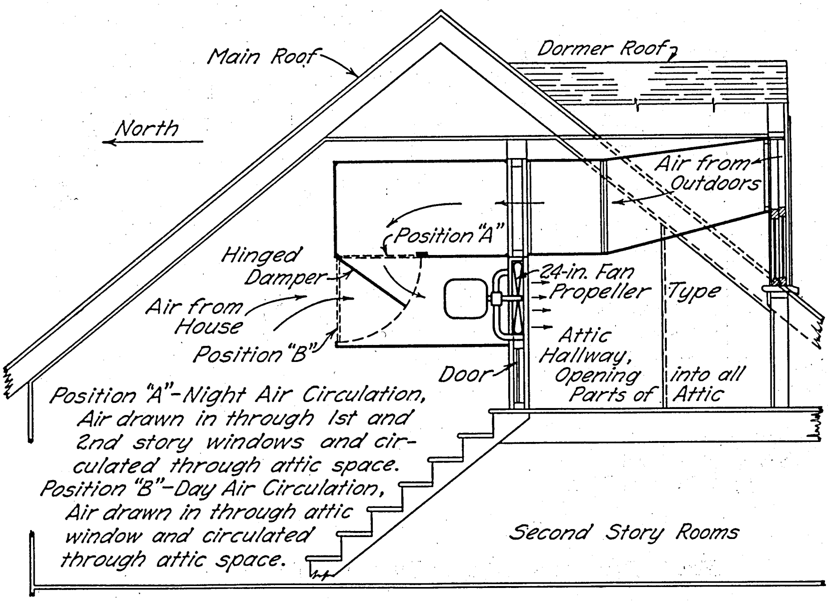

FIG. 4. ARRANGEMENT OF ATTIC FAN IN RESEARCH RESIDENCE

third story during the day. Under both arrangements the fan delivered approximately 4000 cu. ft. of air per minute, which, in terms of the space on the first and second stories, was equivalent to approximately 17 air changes per hour. Based on the space on the second story the delivery was equivalent to about 34 air changes per hour.

Unit Room Coolers

The unit room coolers* and the arrangements for testing them are shown in Fig. 5, and their location in the living room on the first floor of the Research Residence is shown in Fig. 6. Unit *A* was a portable insulated ice chest, consisting of two compartments, one located above the other. The upper compartment was for ice storage, with a maximum capacity of 300 lb. of ice, while the lower compartment formed a tank for collecting the meltage and moisture condensed from the air. A fan circulated the room air through the unit.

Unit *B* included an insulated ice storage tank of 500 lb. capacity, located in the basement, and a cooling unit consisting of an attractively finished cabinet enclosing an extended surface cooling coil, fan, and dehumidification drip pan. The chilled water from the storage

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*See footnote p. 123.

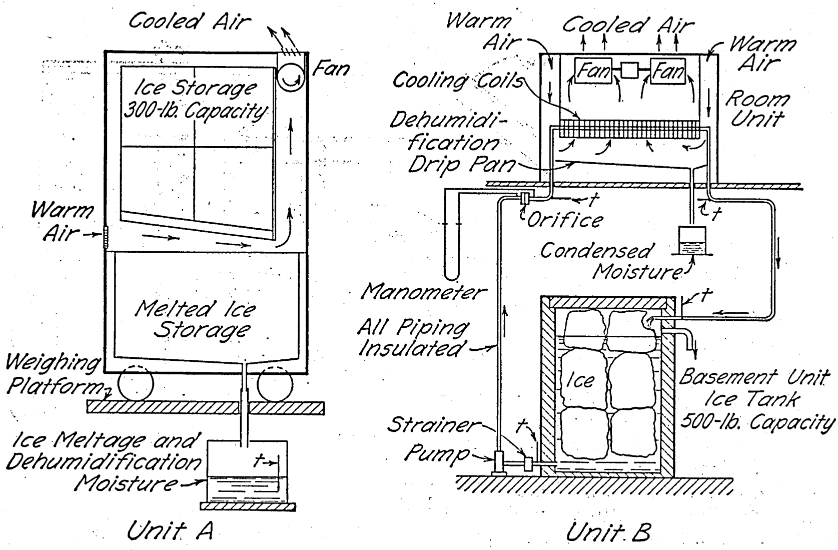

FIG. 5. DIAGRAM OF COOLING UNITS AND TEST ARRANGEMENT

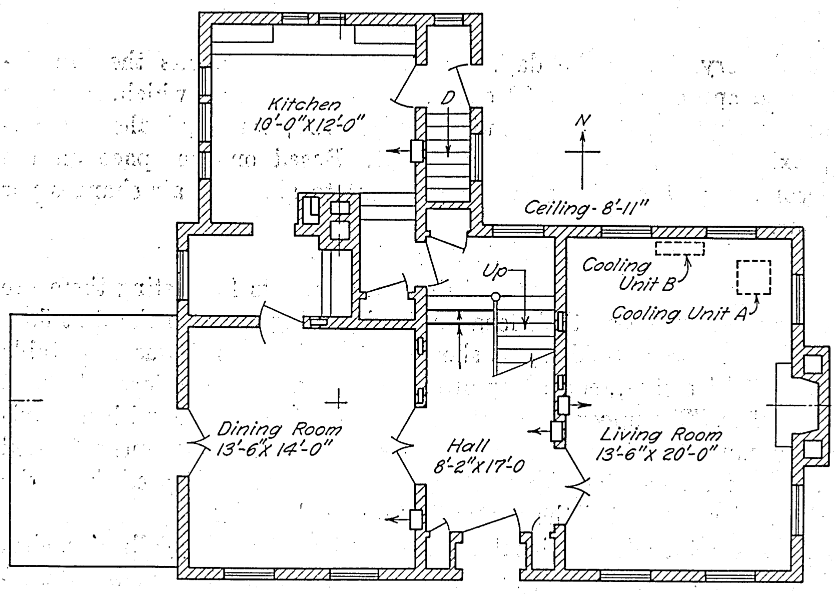

FIG. 6. FIRST-FLOOR PLAN OF RESEARCH RESIDENCE SHOWING LOCATION OF COOLING UNITS

tank in the basement was pumped through the cooling coil, over which warm air from the room was circulated by means of a fan.

The operation of both units was controlled by a room thermostat which served to start and stop the circulating fans, in accordance with the cooling requirements necessary for maintaining a constant air temperature.

2. *Method of Conducting Tests.*—During the summers, from June to October, very complete records were made of the indoor and outdoor air conditions by means of temperature recorders, thermometers, thermocouples, and psychrometers, and many incidental observations were made from time to time. During the periods requiring cooling, detailed records of the operation of the plants, including water and electrical power consumption, and the cooling load, were made at regular intervals both day and night.

Test Procedure with the Plant Using Ice for Cooling

All of the windows, including those in the attic, remained closed continuously day and night, and the cooling plant was started when the effective temperature indoors rose to 75 deg. F. irrespective of the outdoor temperature. This usually occurred when the indoor dry-bulb temperature reached a value between 78 and 82 deg. F., depending upon the relative humidity. After starting the plant the indoor dry-bulb temperature remained practically constant within a range of from 78 to 80 deg. F., but, due to the reduction in the moisture in the air, the indoor effective temperature usually was lowered to approximately 72 deg. F. The operation of the plant was continued until the outdoor dry-bulb temperature was several degrees below that indoors.

Water, chilled with ice in the bunker, entered the cooling coil at about 35 deg. F. A constant quantity of approximately 396 gal. of water per hour was circulated and the average temperature of the water in the coil was about 39 deg. F. Under these operating conditions the temperature of the air at the inlet registers to the rooms did not fall below 60 deg. F. The cooling load on the residence over any interval of time was determined by the amount of ice melted. No air, other than that which leaked in by natural infiltration, was provided for ventilation or cooling at night.

Test Procedures with the Plant Using Mechanical Refrigeration

During all tests, with and without cooling with outdoor air at night, the windows in the attic remained opened and the windows on the first story remained closed. For the purpose of cooling with outdoor air at night, eleven windows on the second story and the door at the head of the stairs leading to the attic were opened.

A. Cooling with Mechanical Refrigeration During the Day and the Circulation of Outdoor Air at Night

At 7 a.m. the fan, which had been delivering outdoor air through the duct system, was stopped, and the windows on the second story and the attic and basement doors were closed. When the temperature at the thermostat on the second story rose to 81 deg. F., the cooling plant, including the condensing unit, and the circulating fan, was started. The fan delivered 5.5 air changes per hour, of which 4.5 were recirculated air, and one was air admitted for ventilation. The condensing unit operated with thermostatic on and off control, and the fan ran continuously, maintaining an average temperature in the residence of about 80 deg. F. In the evening or during the night when the outdoor effective temperature became slightly lower than the indoor effective temperature, the condensing unit was stopped, and the windows on the second story, and the attic and basement doors were opened; and the dampers in the duct system were set so that the fan delivered outdoor air through the house, equivalent to 9.5 air changes per hour, until 7 a.m.

B. Cooling with Mechanical Refrigeration Without the Circulation of Outdoor Air at Night

The second-story windows remained closed during the day and night, and the fan ran continuously delivering through the cooling system the recirculated air and the air admitted for ventilation. The condensing unit operated with thermostatic on and off control, maintaining an average temperature in the residence of about 80 deg. F.

Test Procedure with Plant Using Water from the City Mains

During all of the tests the windows in the attic remained opened day and night, and the windows on the first story remained closed. No tests were made in which artificial cooling during the day was not supplemented with the circulation of air from the outdoors at night. The windows on the second story and the attic and basement doors were closed at 7 a.m., and the damper and door in the duct system changed so that the fan, which had been delivering outdoor air through the house, started delivering recirculated air and air admitted for ventilation. The total quantity of air delivered was approximately 1300 cu. ft. per minute, which was equivalent to 5.5 air changes per hour, and of this amount, 4.5 air changes were recirculated air, and one air change was ventilating air from outdoors. The fan ran continuously during the day and the amount of cooling was regulated by the flow of water through the cooling coil. This was controlled by the thermostatically actuated motor valve located in the outlet water line from the coil. The thermostat located in the hall on the second story was set for 80 deg. F., and the cooling plant was allowed to operate to maintain that condition, until the effective temperature outdoors reached a value equal to the effective temperature indoors. The water valve was then closed, the second story windows, and attic and basement doors opened; and the damper and door in the duct system arranged so that the fan delivered outdoor air through the house at a rate of 2300 cu. ft. of air per minute, or 9.7 air changes per hour. This method of operation was continued until 7 a.m., when the arrangement was again changed for artificial cooling during the day.

Test Procedure for Circulation of Outdoor Air at Night Without Artificial Cooling During the Day

Numerous tests of this type were made and have been reported in detail in a research paper* of the American Society of Heating and Ventilating Engineers, and only several of the most important arrangements will be mentioned here.

A. Circulation of Outdoor Air at Night with Attic Fan

The attic windows remained open and the first-story windows remained closed at all times. At 6 p.m. the second-story windows were opened and the damper at the inlet to the fan was placed in position *A*, as shown in Fig. 4, and the fan started. When in this position the fan delivered air from the first and second stories into the attic whence it passed to the outdoors. The fan was allowed to run until 6 a.m. on the following morning when it was stopped; the damper at the inlet was placed in position *B* and the second story windows were closed. The house remained closed during the day, and no attempt was made to control the indoor temperature.

B. Circulation of Outdoor Air at Night with Basement Fan

Two series of tests using the basement fan or the fan in the main duct system were made with two different arrangements of window openings at night. In one series the windows were fully opened on only the second story, while in the other series they were fully opened on both the first and second stories. The attic windows remained opened at all times. In both cases, at 6 p.m. the basement door, the

*See footnote p. 123.

attic door, and windows on either the second story or on both the first and second stories, were opened. The basement fan was started and during the night it delivered approximately 2200 cu. ft. per min. or 9.3 air changes per hour to the rooms through the duct system. At 6 a.m. the windows and doors were closed, the fan was stopped, and no attempt was made to control the indoor temperature during the day.

Test Procedure with Unit Room Coolers

Both unit room coolers *A* and *B* shown in Figs. 5 and 6, were operated with a thermostat which served to start and stop the air-circulating fans in accordance with the cooling requirements needed to maintain a constant indoor temperature of approximately 80 deg. F. They were started and placed under thermostatic control when the indoor effective temperature reached a predetermined value, and they continued to operate until the outdoor effective temperature became equal to the indoor effective temperature, when they were stopped and the windows were opened.

Unit cooler *A*, Fig. 5, was mounted on a portable platform scale, and the ice melting and dehumidification rates were determined by direct weighing. This afforded a very convenient method of determining the cooling and dehumidifying capacities of the unit. The majority of the tests with this unit were made in the living room of the Residence in the location shown in Fig. 6, and although the direct cooling was confined to the one room by closing the door leading to the hall, complete records were made of the air conditions in all of the rooms.

In addition to the air-circulating fan in the cabinet containing the extended surface cooling coil, the operation of cooling unit *B* required the running of a pump which circulated chilled water from the insulated ice tank in the basement to, and through, the cooling coil in the cabinet located in the living room. The pump ran continuously during the periods requiring cooling. The total cooling load or heat absorbed from the air was determined by the ice meltage, and the amount of dehumidification or moisture condensed from the air was determined by collecting and weighing the water which drained from the cooling coil. During most of the tests made with this unit the door from the living room to the interconnecting hall remained open, and the unit was allowed to cool the entire first story, consisting of the living room, hall, dining room, breakfast room, and kitchen, having a total net volume of 7300 cu. ft. Complete records were made of the air conditions in all of the rooms in the house.

3. *Results of Tests.*—It is possible to include only the most general and practical results of the tests made during the four summers. The complete results have been given in the research papers* of the American Society of Heating and Ventilating Engineers previously mentioned.

Results of Tests with the Plant Using Ice for Cooling

The results indicated that, with minor changes and adjustments, a central forced air heating system could be successfully used to distribute cooled air during the summer months. By means of dampers in the ducts the distribution of cooled air was adjusted to balance the cooling load in the various rooms, and when balanced, the rooms remained uniformly cooled during the entire day and night, irrespective of the position of the sun or the change in the cooling load. The modulating damper actuated by the single thermostat located in the dining room and which served to proportion the amount of return air passing through the cooling coil in accordance with the cooling load, gave very excellent regulation of the indoor air temperature. The cool air was delivered from the registers at temperatures varying from 60 to 70 deg. F., depending upon the cooling load and at velocities varying from 50 to 450 ft. per minute. Although the cooled air was delivered to all of the rooms, except one, through baseboard registers, the average temperature difference between the floor and the ceiling was only about 3.5 deg. F. An objectionable draft was noticeable in only one room, and in that case only in the proximity of the register where the air was being delivered at a velocity of approximately 450 ft. per minute. A baffle placed in front of the register to direct the air toward the ceiling easily corrected this condition. The average rise in the temperature of the air from the fan inlet to the register faces was about 3 deg. F., and although no insulation was used on any of the duct surfaces, except the section directly surrounding the cooling coil, no condensation appeared on the ducts or furnace casing.

The chilled water entered the cooling coil at a temperature of 35 deg. F. and the mean temperature of the water in the coil varied from 39 to 44 deg. F., depending upon the cooling load. This temperature was considerably below the dewpoint temperature of the air entering the coil, and, as a result, a comparatively large amount of moisture was condensed from the air as it passed through the coil. The largest amounts of dehumidification occurred during the first hour or

*See footnote p. 123.

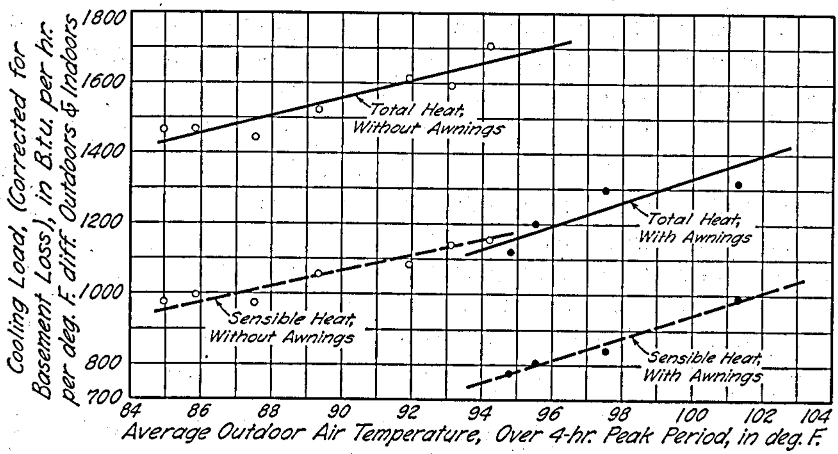

FIG. 7. COOLING LOAD ON RESEARCH RESIDENCE WITH AND WITHOUT AWNINGS USING ICE REFRIGERATION

two following the starting of the plant, and the rate usually decreased until, after several hours of operation, it became fairly constant. Data showed decreases from as high as 9 lb. per hour to as low as 4.5 lb. per hour within two hours after cooling was started. The reduction in the indoor relative humidity to a stabilized value of about 45 per cent reduced the effective temperature from the starting condition of 75 deg. to an operating condition of 72 deg. The average dry-bulb temperature after several hours of operation was usually about 78 deg. F. The indications were that these conditions were slightly cooler than are necessary for the average residence cooling installation.

The reduction in the cooling load due to the application of awnings to all of the east, south, and west windows is shown in Fig. 7. These curves are based on the averages of data taken during the four hours of the day when the outdoor temperature was the highest. The total cooling loads with and without the awnings on the windows are represented by the solid lines, and the reduction in the load effected by the awnings was of the order of 30 per cent.

With the house closed during the day and night, and no provision made for ventilating air except that which leaked in by natural infiltration, objectionable odors were noticeable on many days.

Results of Tests with the Plant Using Mechanical Refrigeration

In using outdoor air at night to cool a residence, either alone or in conjunction with artificial cooling during the day, it was considered undesirable to leave the windows on the first story open all night. For

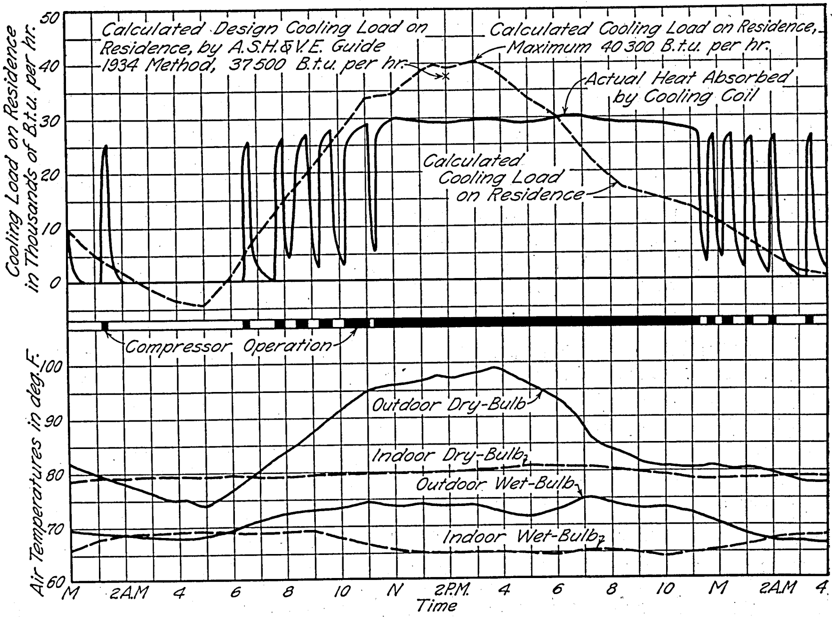

FIG. 8. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES, TEST NO. 7-34, JUNE 27, 1934; NO NIGHT AIR COOLING, USING MECHANICAL REFRIGERATION

this reason the windows on the first story remained closed at all times during the summer of 1934, irrespective of whether cooling with outdoor air at night was or was not used to supplement the artificial cooling during the day.

The operating characteristics of the mechanical refrigerating plant using artificial cooling during the day and the circulation of outdoor air at night were similar to those shown in Fig. 8, with the following exceptions: The beginning of the operating period was usually several hours later than 6 a.m., the time at which the windows on the second story and the attic door were closed and the duct system changed over to recirculating indoor air and ventilating air. The duration of the continuous operating period was much shorter, and the end of the intermittent operating period, when the windows and attic door were opened and the duct system was changed over to circulating night air was usually before 10:00 a.m.

Figure 8 shows the actual and calculated cooling load on the residence, and the operating characteristics of the plant during a typical 24-hour period when no cooling was done with outdoor air at night. The circulating fan continued to run the entire period, circulating indoor air and ventilating air through the duct systems. The compressor or condensing unit was under thermostatic control. At 6:30 a.m. on this particular day the indoor temperature on the second floor had risen to 80 deg. F. and the condensing unit was started through the action of the thermostat. From 6:30 a.m. the condensing unit operated intermittently until 11:20 a.m., and from then, until 11:15 p.m., the operation was continuous at full load capacity. At 11:15 p.m. the intermittent operation was resumed. The outdoor temperature was 97 deg. F. at 2:00 p.m. and it reached a maximum of 99 deg. F. at 3:45 p.m. The average indoor dry-bulb temperature varied from 79 deg F. at 8:00 a.m., to about 81 deg. F. at 5:00 p.m., and the indoor relative humidities and effective temperatures at these times were 60 per cent and 74.5 deg., and 42 per cent and 73.5 deg. During the period of continuous operation the average indoor relative humidity was about 45 per cent, and the average indoor effective temperature was about 73.5 deg. The indications were that these conditions are conducive to comfort in the average residence cooling installation where the exposure period is of several hours duration and the degree of activity is moderate. Also, that the introduction of outdoor air equivalent to one air change for the purpose of ventilation was sufficient to prevent objectionable odors.

Again referring to Fig. 8, it may be noted that from 6:00 a.m. to 10:00 a.m. the actual average load approximated the calculated load, and that from 10:00 a.m. to 6:10 p.m. the actual load was less than the calculated load. At 6:10 p.m. the two loads became equal. The actual load never exceeded 30 500 B.t.u. per hour, while the calculated load attained a maximum of 40 300 B.t.u. per hour at 3:00 p.m. The effect of the heat lag and the heat storage capacity of the structure is best shown in the period from 6:10 p.m. to 11:15 p.m. During this period the condensing unit continued to operate at approximately full capacity, although the calculated load had decreased rapidly, until at 11:15 p.m. it was less than one-half of the capacity of the unit.

Figure 9 shows the relationship between the degree-hours above 85 deg. F. per day, the hours of compressor operation, and the total and sensible heat absorbed by the cooling coil. The number of degree-hours per day above 85 deg. F. is an index of the severity of the day, and is calculated by taking into consideration the number of hours during the day that the outside temperature is above 85 deg. F., and the number of degrees by which the average temperature during this period exceeds 85 deg. F. For example, if on a given day the outside temperature is 90 deg. F. for 6 hours during the day, and during all

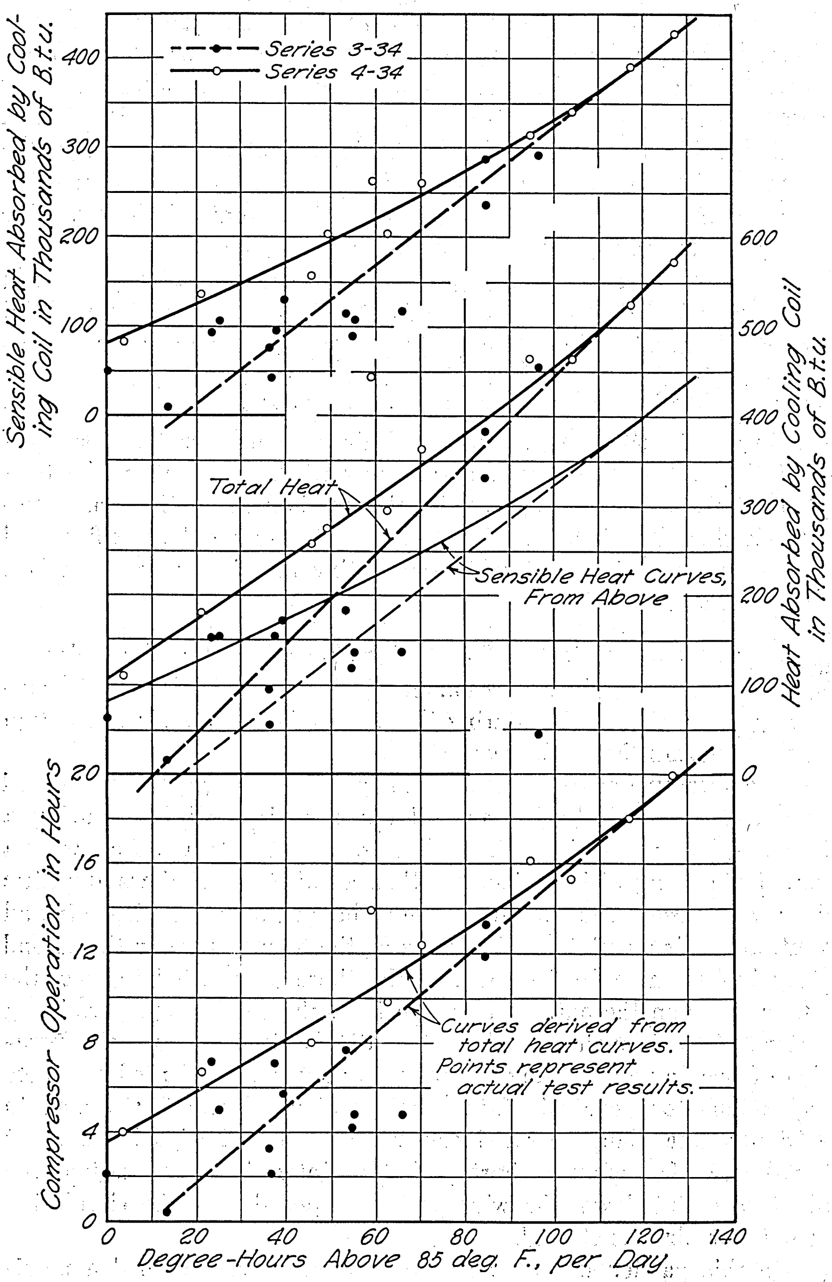

FIG. 9. HEAT ABSORBED BY COOLING COILS AND HOURS OF COMPRESSOR OPERATION PER DAY

the remainder of the time it is below 85 deg. F., then the number of degree-hours for that day is the product of 6 and 5, or 30. The base temperature of 85 deg. F. was chosen because it was found that usually no cooling was required unless the outside temperature was 85 deg. F. or above. The degree-hour in cooling is somewhat analogous to the degree-day in heating, and it affords a means of comparing the severity of days and seasons from the standpoint of the probable amount of cooling that would have been required.

The two lower curves in Fig. 9 indicate the amount of saving in compressor operation that may be effected by supplementing artificial cooling during the day with cooling with outdoor air at night, as provided in the Research Residence, where only the second-story windows and attic were opened and 9.5 air changes per hour were provided with the fan in the forced-air system. For the season of 1934 the indications were that a saving of 20 per cent in the time of plant operation was possible by this method of ventilation. However, it should be kept in mind that in determining the net financial saving the cost of operating the ventilating fan should be taken into consideration.

Results of Tests with the Plant Using Water from the City Mains

The comparatively mild summer of 1935 made it possible to supplement artificial cooling during the day with outdoor air circulation at night during the entire season.

The operating characteristics of the plant, the actual and calculated cooling loads, and the indoor and outdoor air conditions during a typical 24-hour period are shown in Fig. 10. At 7:00 a.m., following a period of cooling with outdoor air, the second-story windows and attic door were closed, and the duct system changed over to indoor air and ventilating air circulation. At 8:50 a.m. the temperature on the second story had risen to 80 deg. F., and the thermostat acted to open the motor valve on the outlet water line from the cooling coil permitting water to flow through it. This comparatively simple automatic operation of opening and closing the water supply valve afforded complete and satisfactory control of the cooling plant. From 11:15 a.m. to 9:25 p.m. the plant operated continuously at full capacity, and at 11:20 p.m., following a short period of intermittent operation, the plant was shut down, the windows were opened, and the circulation of outdoor air was started.

On this particular day the outdoor temperature reached a maximum of 96 deg. F. at 2:30 p.m. From about 11:00 a.m. to 6:30 p.m. the calculated load exceeded the actual load, while from 6:30 p.m. to

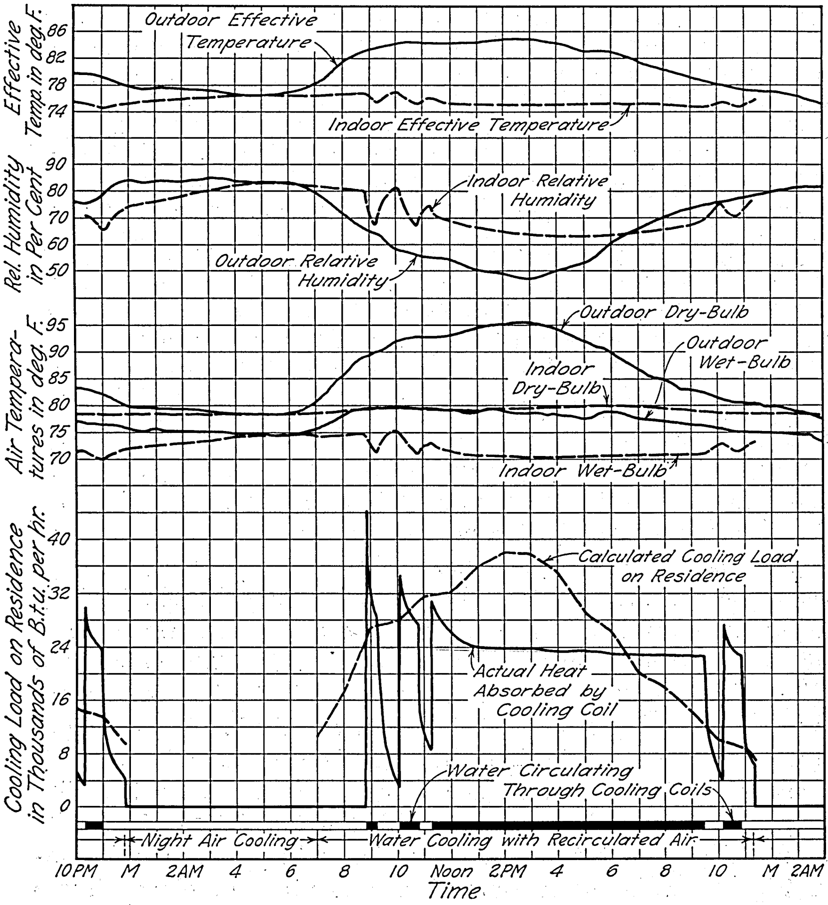

FIG. 10. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES, TEST NO. 13, SERIES 1-35, AUGUST 2, 1935; WITH NIGHT AIR COOLING, COOLING PLANT USING WATER FROM CITY SUPPLY MAINS

9:25 p.m. the calculated load was less than the actual load. The maximum calculated load of about 37 500 B.t.u. per hour occurred between 2:00 and 3:00 p.m., and the actual load at this time was about 24 000 B.t.u. per hour, which, under normal operating conditions, was the approximate capacity of the plant.

The indoor effective temperature at 8:50 a.m. when the plant started to operate was 76.5 deg. F. This condition, which certainly could not be considered a comfortable one, was due to the unusually high indoor relative humidity of 80 per cent. Although the plant operated to maintain an indoor dry-bulb temperature of 80 deg., the unsatisfactory condition was not materially improved until noon, and even then, and through the period of continuous operation, the lowest indoor effective temperature attained was 75 deg. The minimum indoor relative humidity during this period was 63 per cent. The results indicated that on days when the outdoor temperature exceeded 90 deg. F. for a major portion of the day and the cooling plant operated continuously, an indoor effective temperature of 75 deg. could be maintained with 58 deg. F. water. While this could not be regarded as an optimum condition for comfort it was a material improvement over that which would have existed without any cooling.

On mild days when the outdoor temperature did not exceed 90 deg. F., the plant operated intermittently and the indoor relative humidity was always above 63 per cent. With this relative humidity the maintenance of an indoor dry-bulb temperature of 80 deg. F. resulted in an effective temperature above 75 deg. F., which was entirely unsatisfactory. This condition could be improved somewhat by operating the cooling plant more often or continuously, and an effective temperature not exceeding 74 deg. could be obtained by maintaining a dry-bulb temperature below 80 deg. F. Under these conditions the relative humidities were usually from 70 to 80 per cent.

From these results it may be seen that the limitation of the plant using water at a temperature of 58 deg. F. was the inability to maintain reasonably low relative humidities. Although a material quantity of moisture was removed from the air, the minimum relative humidity of 63 per cent obtained was relatively high in comparison with the 45 per cent obtained with the plants using ice and mechanical refrigeration. The rise in the temperature of the water in the cooling coil was 8 deg. F., giving a mean water temperature of 62 deg. F. The resistance of the coil to the flow of water was negligible, but the pressure loss in the line between the city main and the coil was very large, and in some installations it might be a limiting factor in determining the amount of water that could be circulated through the coil.

Results of Tests with the Circulation of Outdoor Air at Night

A number of tests with various arrangements of window openings and fan conbinations were made and reported in a previous paper.* In that paper it was shown that the minimum indoor temperature attained with the circulation of outdoor air at night was at about 6:00 a.m. when the windows were closed. Furthermore, it was shown that the effectiveness of different methods of cooling with outdoor air

*See footnote p. 123.

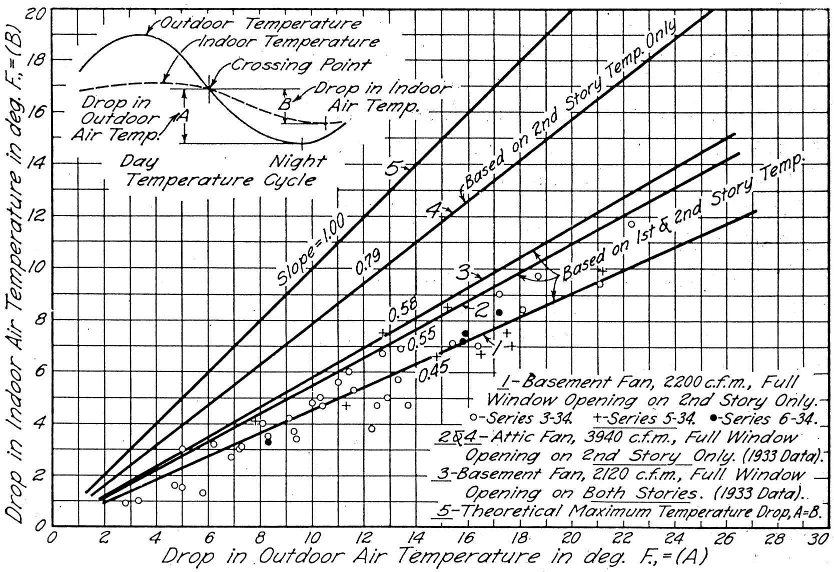

FIG. 11. TEMPERATURE DROPS FOR VARIOUS METHODS OF NIGHT AIR COOLING IN RESEARCH RESIDENCE

at night could be compared by plotting the drops in indoor air temperatures from the time that the indoor and outdoor temperatures became identical to the time at which the indoor temperature attained a minimum, against the drop in outdoor temperature from the time at which the indoor and outdoor temperatures became identical to the time at which the outdoor temperature attained a minimum. These temperature drops are shown as *B* and *A* respectively in the insert in Fig. 11, which gives the temperature drop curves for several different fan and window combinations. The maximum possible effectiveness would be representative of a condition where the minimum indoor air temperature was the same as the minimum outdoor condition, and this is represented in Fig. 11, by Curve No. 5, a 45-degree line having a slope of 1.0 when the two temperature drops are plotted to the same scale. Curves Nos. 1 and 3 with respective slopes of 0.45 and 0.58 show the temperature drops based on the average temperatures on both stories, when the fan in the forced air system was circulating the outdoor air equivalent of 9.5 air changes per hour. The difference in the slopes of the two curves was due to the difference in the arrangement of window openings, Curve No. 3 representing the condition attained with full window opening on both the first and second stories, and Curve No. 1 representing that attained with full window opening on the second story only. Curves Nos. 2 and 4 show the temperature drops for outdoor air circulations with the attic fan equivalent to 17 air changes per hour based on the two stories, and 33.5 air changes per hour based on the second story alone. Curve No. 2 represents the average condition attained on both the first and second stories with full window opening on both stories, while Curve No. 4 represents the condition attained on the second story with only the windows on that story opened.

Considering that in most communities it is undesirable to allow first-story windows to remain open the entire night, the most practical methods of operation may be limited to those represented by Curves Nos. 1 and 2 with windows opened on the second floor only, and the results based on the cooling attained on both stories. A comparison of these two curves shows that in cooling the house as a whole the attic fan was but slightly more effective than the basement fan, even though the quantity of air which it delivered was 80 per cent greater.

Although the temperature drop curve obtained by natural ventilation with full window opening on all stories is not shown in Fig. 11, it is interesting to note that the position of this curve, if included, would be slightly above Curve No. 3, indicating that very effective cooling was attained in the Research Residence without any fan, if the entire house remained opened during the night. However, it is very probable that this residence, with the full third story and its generous number of windows on all stories, was better adapted to natural ventilation than the average house would be.

Experience indicated it was reasonable to consider that the effectiveness of cooling with outdoor air at night was dependent upon the quantity of air circulated, rather than upon the method used, and that the slopes of temperature drop curves as shown in Fig. 11 were legitimate measures of the effectiveness of the different methods. Under these considerations the slopes of all the temperature drop curves obtained when the outdoor air was positively circulated by means of some type of fan are shown in Fig. 12, plotted against the respective quantities of air circulated expressed in air changes per hour. The upper curve in this figure is based on full window opening and average indoor air temperatures attained in the entire space under consideration. In a two-story house it represents the average indoor conditions attained in the entire house with full window openings on both stories, or the average indoor conditions attained on the second story with full window openings on the second story only. In each

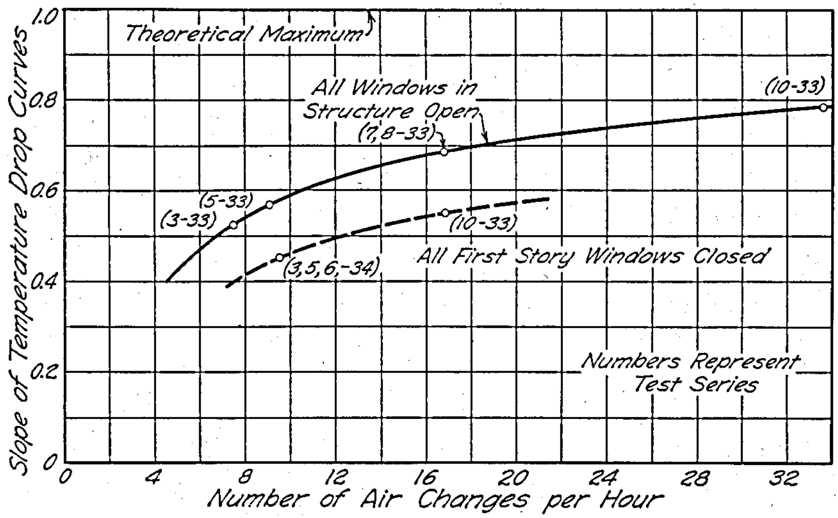

FIG. 12. INFLUENCE OF RATE OF CIRCULATION OF NIGHT AIR ON COOLING EFFECT IN RESEARCH RESIDENCE

case the number of air changes per hour was based on the space in which the windows remained opened. The lower curve in Fig. 12 represents conditions in a two-story house in which the second-story windows only are opened, and the effectiveness of cooling and the number of air changes per hour are based on the entire house.

The curvature of the curves in Fig. 12 shows that for the smaller number of air changes per hour the cooling effected increases rather rapidly with increases in the amount of air circulated up to about ten air changes per hour. Neglecting the cooling effect of air motion the curves further indicate that effective cooling with the circulation of outdoor air at night cannot be accomplished with less than ten air changes per hour, and that above thirty air changes per hour the gain resulting from increasing the amount of air circulated is very small.

Results of Tests with Unit Room Coolers

There were no means available for varying the cooling capacity of the portable unit room cooler *A*, except by starting and stopping the fan which delivered the room air through the unit. This was done by a room thermostat mounted on a support near the center of the room, and the fluctuation in dry-bulb temperature between the on and off periods did not exceed 1.5 deg. F. The average ice melting rate or cooling capacity of the unit when operating intermittently in a room with a dry-bulb temperature between 75 and 76 deg. F. was about 15 lb., or 2400 B.t.u. per hour. The average drop in the air temperature through the unit was approximately 11 deg. F. and the reduction in the moisture content of the air was of the order of 10.5 grains per lb. of dry air. The relative humidity maintained in the room was between 50 and 55 per cent as compared with that of 65 to 70 per cent in the adjoining hall, which was not cooled. The tests with this unit were limited in number, and were made during mild weather when the outdoor temperature did not exceed 90 deg. F. They are, therefore, only indicative of the operating characteristics of the unit, and the cooling load mentioned should not be accepted as that of an average size room in a residence during severe summer weather.

By changing the amount of chilled water circulated through the cooling coil by throttling, or the quantity of air delivered by the fan through the use of the three speed control provided with it, the cooling capacity of room cooler *B* could be varied over a rather large range. In cooling the entire first story of the residence, including the living room in which the unit and control thermostat were located, the hall, dining room, breakfast room, and kitchen, the maximum amount of chilled water was circulated and the fan was operated on low speed. On a typical day when the outdoor temperature reached a maximum value of 94 deg. F. at about 3:30 p.m., the unit operated intermittently for a period of 11.7 hours, from 1.20 p.m. to 1:00 a.m. on the following morning. During that time it was possible to maintain indoor temperatures of 79.3 deg. F. in the living room, 81.5 deg. F. in the hall, 82.6 deg. F. in the dining room, and 82.1 deg. F. in the kitchen, or an average of 81.4 deg. F. for the entire first story. The maximum indoor temperature on the uncooled second story was 87.5 deg. F. at 7:30 p.m. Although the cooling unit and control thermostat were unfavorably located with respect to the center of the first story, the air temperatures in any one room did not deviate more than 2.1 deg. F. from the mean temperature for the entire first story. The minimum temperature of 79.3 deg. F. occurred in the living room where the cooling unit was located and the maximum temperature of 82.6 deg. F. occurred in the dining room. The average initial relative humidity of approximately 68 per cent was reduced to a value of 58 per cent at the end of one hour's operation, and at the end of five hours' operation it had attained a stabilized value of about 50 per cent. The average rate of heat absorption reduced to an equivalent ice melting capacity and expressed as tons of refrigeration was 0.59 tons. In general, the performance of the unit and the conditions attained on

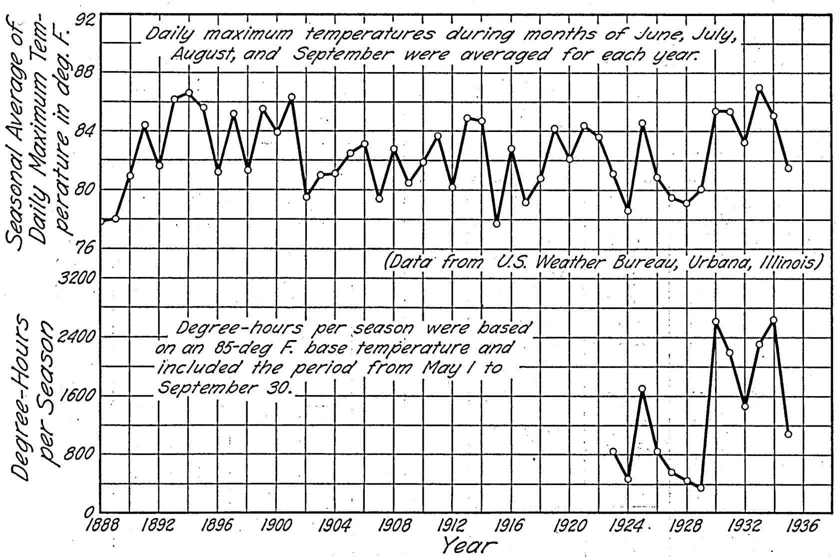

FIG. 13. GRAPHICAL RECORD OF SUMMER TEMPERATURES AT URBANA, ILLINOIS, FROM 1888 TO 1935

the entire first story were satisfactory, and undoubtedly the variation between rooms would have been improved by a more centralized location of the unit and control thermostat.

4. *Seasonal Weather Data and Operating Costs.*—The wide variation in the weather during the cooling seasons of 1932, 1934, and 1935, and the difference in the methods of operation and the indoor conditions maintained with the three central cooling plants, makes any comparison of seasonal operation costs very difficult. In considering operating costs it should be kept in mind that they are directly dependent upon the power, water, and ice rates in the particular community under consideration, as well as upon the type and efficiency of the plant, and, therefore, data secured in one community, such as Urbana, Illinois, should, when applied to another community, be adjusted for the variation in the utility rates.

Figure 13 shows a graphical record of the summer temperatures in Urbana, Illinois, from 1888 to 1935. The daily maximum temperatures during the months of June, July, August, and September, were averaged for each year, and these values constitute the seasonal averages of daily maximum temperature that are shown in the upper curve. The data were secured from the local station of the U. S. Weather Bureau.

TABLE 1

SEASONAL WEATHER DATA AND HOURLY OPERATING DATA FOR THE CENTRAL COOLING PLANTS

Season	Type of Cooling Plant	Number of Days With Temperature Above 85 deg. F.	Total Degree-Hours Above 85 deg. F. Base	Indoor Conditions Maintained			Cooling Capacity of Plant B.t.u. per Hour*
				Dry-bulb Temperature, deg. F.	Relative Humidity per cent	Effective Temperature, deg. F.	
1932	Ice......................	62	1470	78.0	45	72.0	31 700
1934	Mechanical...............	65	2660	80.0	45	73.5	29 000
1935	Water from City Mains.....	49	1180	80.0	65	75.5	24 000

TABLE 1 (CONCLUDED)

Season	Type of Cooling Plant	Operating Costs, Cents per Hour†						Total Cost per Hr. per Ton of Refrigeration Delivered, cents	Cost of Operating Fan for Night Air Cooling, cents per hr.
		Ice	Water	Power for			Total		
				Circulating Pump	Compressor	Recirculating Fan			
1932	Ice....................	44.00	.....	1.55	.....	0.82	46.37	17.55	....
1934	Mechanical.............	.....	3.17	.....	9.05	1.05	13.27	5.49	1.27
1935	Water from City Mains....	.....	11.88	.....	.....	1.34	13.22	6.61	1.37

*After several hours of continuous operation.

†Based on the following rates: ice, $4.00 per ton; water, 33 cents per 1000 gal.; power, 3.1 cents per kw.-hr.; no capital costs taken into consideration.

The lower curve in Fig. 13 shows the degree-hours per season from 1923 to 1935. The degree-hours per season were based on an 85 deg. F. base temperature and included the five months of May, June, July, August, and September, and from these data it may be observed that the degree-hours for the seasons of 1932, 1934, and 1935 were 1470, 2660, and 1180 respectively. The degree hours for the season of 1933, which was devoted to numerous incidental tests involving cooling with different methods of circulating outdoor air at night, the operation of the central cooling plant with limited amounts of ice, and tests with the unit room coolers, were 2310. Thus it is obvious that seasonal operating costs cannot be compared directly, due to the great variation in the severity of the seasons.

Furthermore, the data indicate the futility of attempting to predict or estimate the severity of any approaching season from a knowledge of the preceding ones. During the twelve-year period from 1923 to 1935, the mean number of degree-hours per season was 1350, the minimum during any season was 360 degree-hours, and the maximum was 2660 degree-hours. Using the number of degree-hours per season as a legitimate measure of the severity of the seasons, these data indicate a fluctuation of 750 per cent between the mildest and the severest season, and a variation from the mean of as much as 350 per cent.

Granting the limitations of the applicability of operating costs determined in only one given community and in one residence, Tables 1 and 2 are included as a matter of general information, to be used with discretion by experienced air conditioning engineers capable of appreciating the limitations involved. No account of the initial costs of the plants has been taken into consideration.

Table 1 gives seasonal weather data and hourly operating data for the three types of central cooling plants operated in the Research Residence during the summers of 1932, 1934, and 1935, using ice, mechanical refrigeration and water from the city water mains for cooling and dehumidifying the air. The operating costs are based on the actual measured ice, water, and power consumptions of the various units included in the systems, and when the plants were operating at normal capacity under approximately equilibrium conditions. The local water and power rates of 33 cents per thousand gallons, and 3.1 cents per kw.hr., were used in the calculations, and the price of ice was assumed to be $4.00 per ton. In comparing the total operating cost per hour with the different cooling plants, cognizance must be taken of the variation in the capacities of the plants

TABLE 2
SEASONAL OPERATING COSTS* (CALCULATED) FOR THE CENTRAL COOLING PLANTS

Season	Type of Cooling Plant	Number of Days with Temperature Above 85 deg. F.	Total Degree-Hours Above 85 deg. F. Base	Operating Conditions			
				Awnings		Circulation of Outdoor Air at Night	
				With	Without	With	Without
1932	Ice	62	1470	Yes	....	....	Yes
				....	Yes	....	Yes
1934	Mechanical	65	2660	Yes	....	Yes	....
				Yes	....	....	Yes
1935	Water from City Mains	49	1180	Yes	....	Yes	....

TABLE 2 (CONCLUDED)

Season	Type of Cooling Plant	Seasonal Operating Costs in Dollars (Calculated)†							Ratio of Minimum to Maximum Seasonal Cost	Per Cent Reduction in Seasonal Operating Costs
		Ice	Water	Power for Operating						
				Circulating Pump	Compressor	Fans		Total		
						Recircu-lating	Outdoor Air			
1932	Ice	161.20	.....	8.72	.....	4.64	.....	174.56	0.678	32.2
		238.80	.....	12.20	.....	6.47	.....	257.47		
1934	Mechanical	.....	12.89	.....	31.87	5.86	12.03	62.65	0.834	16.6
		.....	19.10	.....	47.28	8.71	.....	75.09		
1935	Water from City Mains	.....	16.83	.....	.....	10.37	9.71	36.91		

*Assuming that the plants were operated the entire season under the conditions indicated.
†Based on the following rates: ice, $4.00 per ton; water, 33 cents per 1000 gal.; power, 3.1 cents per kw.-hr.; no capital costs taken into consideration.

and the indoor conditions maintained in the residence. Probably the most legitimate comparison can be made from the values given in the column "total cost of operation per ton of refrigeration delivered in cents per hour." These values are based on the total amount of heat, including the sensible and the latent heat, removed from the air as it passed through the cooling coils. With the plants as installed and operated these values were 17.55 cents for the plant using ice, 5.49 cents for the mechanical refrigeration plant, and 6.61 cents for the plant using water from the city water mains.

Table 2 is included to give some approximate practical information as to what would have been the seasonal cost for cooling the residence under the conditions, and during the seasons indicated. Due to the variations in the severity of the seasons, in the operating conditions, and in the indoor conditions maintained, comparisons should not be made between the seasonal operating costs for different summers.

Actually, with exception of the summer of 1935, the residence was not cooled by any one method during an entire season. In 1932 one series of tests was conducted with the windows of the residence equipped with awnings and another series was conducted without awnings. The total seasonal operating costs in dollars for the summer of 1932 were calculated, first, on the assumption that the awnings were in place the entire season, and second, on the assumption that no awnings were in place at any time. These calculations were made possible by determining performance curves for each method of operation similar to the lower set of curves shown in Fig. 9, which give the relationship between the degree-hours above 85 deg. F. per day and the compressor operation or plant operation in hours. From curves of this type for each method of operation, and from weather data giving the number of days and degree-hours per day above 85 deg. F. it was possible to estimate the total number of hours that the plant would have had to operate for the entire season. Then, applying the hourly operating costs as given in Table 1, the total seasonal cost was determined. By this method of calculation it was estimated that if the east, south, and west windows of the residence had been equipped with awnings during the entire season, the seasonal operating costs using ice at $4.00 a ton would have been about $174.56, while if no awnings were used the cost would have been $257.47. This indicates that awnings would have effected a saving of approximately $83.00 or 32.2 per cent in the total seasonal operating costs during the summer of 1932 when ice was used as the cooling medium.

During the summer of 1934 when the mechanical refrigerating plant was used, one series of tests was conducted taking advantage of cooling by circulating outdoor air through the house at night whenever it was feasible to do so, and another series was conducted keeping the house closed and using only mechanical refrigeration. Awnings were fitted to all east, south, and west windows the entire summer. Performance curves for the two methods of operation were determined, and from these curves, weather data for the summer of 1934, and the hourly operating costs as given in Table 1, the total estimated seasonal operating costs for cooling the residence during the summer of 1934 with the mechanical refrigerating plant supplemented with the circulation of outdoor air at night was $62.65. If no outdoor air at night was used to supplement the cooling with the mechanical refrigeration plant, the estimated seasonal operating cost would have been $75.09. This indicates that by supplementary cooling with the circulation of outdoor air at night a possible net saving of $12.44 or 16.6 per cent, might have been effected over cooling with mechanical refrigeration alone.

In 1935 only one method of operating the central cooling plant when using water from the city water mains was employed. The windows were equipped with awnings and the mild summer made it possible to supplement the artificial cooling each day with the circulation of outdoor air at night. For this particular season and with the method of operation employed, the total seasonal operating costs were $36.91. It should be remembered, as shown in Table 1, that while the indoor conditions maintained were a substantial improvement over what they would have been without any cooling, they were not as satisfactory as those maintained with the plants used during the summers of 1932 and 1934. The cost items in Tables 1 and 2, other than those specifically mentioned, show the operating costs of the different units included in the cooling systems.

This work is to be continued and during the summer of 1936 tests are to be made using water as the cooling medium at temperatures of 52 deg. F. and 45 deg. F.

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